Control of ventilation and air conditioning plants

Building Technologies

SIEMENS

Contents

1. Temperature control in air treatment systems

1.1 Internal heat sources

If a room is to be kept at a constant temperature, internal heat sources represent disturbances which must be compensated with a large amount of cooling energy during the cooling season. During the entire heating season, however, internal heat sources can be used as heating energy. The following three heat sources can be effective from the point of view of control:

- Heat output of radiators (base load heating)
- Heat supply of the ventilation system
- Internal heat sources (people, devices, lighting, solar radiation etc.)

If the heat supply is properly divided among the base load heating on the one hand and the ventilation system on the other – while paying careful attention to the internal heat sources – there should be no major control problems.

1.2 Supply air temperature control

The simple ventilation system shown in Fig 1-1 has the task of supplying outside air to a room without internal heat gains, while the actual room heating is provided by radiators. In order for the room temperature to remain constant, e.g. at 21 °C, when the outside temperature is at its lowest, the radiators must provide 100 % of the heating load, i.e. cover all heat losses, and the supply air must be blown into the room at 21 °C. In this case, the ventilation system does not contribute to the room heating. Its sole task is to supply fresh air, which it must heat up to the room temperature.

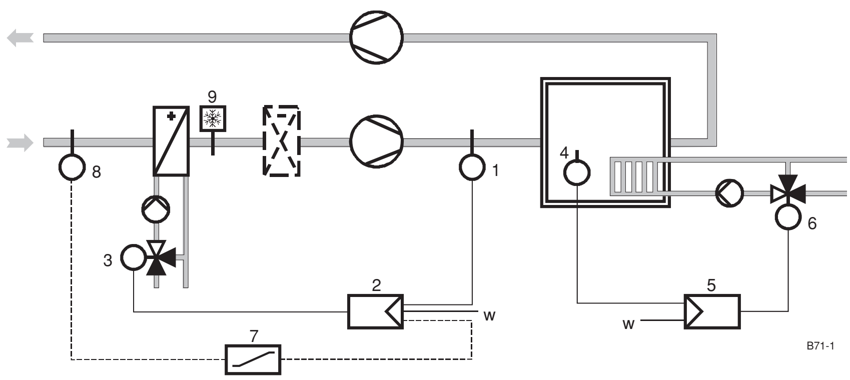


Fig. 1-1 Supply air temperature control with base load heating
1 Supply air temperature sensor
2 Supply air temperature controller
3 Motorized heating valve
4 Room temperature sensor
5 Room temperature controller
6 Motorized heating valve
7 Supply air temperature shift controller
8 Outdoor temperature sensor
9 Frost protection thermostat

The overall plant is controlled as follows:

- So-called supply air temperature control is selected for the ventilation system, i.e. the temperature sensor of the ventilation system is installed in the supply air duct. The modulating temperature controller (2) compares the temperature acquired by the sensor (1) against the setpoint. In case of a deviation, the controller adjusts the heating valve (3) so that a constant supply air temperature is achieved. The supply air temperature setpoint must only be set to the same value as the room temperature if there are no internal heat gains. Otherwise, the supply air temperature should be set to an appropriately lower value, but not so low that drafts occur. The output of the air heating coil must be sized such that the outside air volume required for ventilation can still be heated to the selected setpoint when the outside temperature is at its lowest
- Since any internal heat gain occurring in the room can only be detected using a room temperature sensor (4), room temperature control must be provided for the radiator heating system
- If the supply air temperature is set lower than the room temperature because of the internal heat gains, the room temperature will drop in the absence of the internal heat sources. In this case, the radiator heating system must also be able to heat up the cooler supply air to the desired room temperature in addition to the actual heating load

This type of control must only by used for systems with relatively small internal heat gains. In case of large and irregular internal heat gains in the room, the room temperature controller (5) reduces the radiator temperature to a correspondingly great degree each time. This can often give rise to a more or less on/off operation of the radiators with corresponding temperature fluctuations in the room. Additionally, if the radiators are cold, the curtain of warm air in front of the windows is lost, causing a very uncomfortable state (drafts due to downward flow of cold air at the windows).

If cooling in summer is also required, no guarantees can be made regarding the room temperature. In this case, it is usually advantageous to compensate the supply air temperature setpoint based on the outside temperature (see 1.7) using a shift controller (7).
Supply air temperature control is used where the primary task of the ventilation system is to supply fresh air:

- The primary purpose of ventilation in kitchens, restaurants, garages, sports halls etc is to replace bad room air. The room temperature itself plays a secondary role
- In other rooms, such as workshops, factory buildings, storerooms, etc., supply air temperature control can be used to keep the room temperature constant during the heating season, provided the room contains no variable heat sources but has a 100 % controllable base load heating system. The base load heating system (e.g. radiator system) must be controlled according to the room temperature
- For better adjustment of the supply air temperature to the heat losses of the conditioned space, the supply air temperature can be compensated according to the outside temperature

1.3 Room temperature control

If a relatively large internal heat quantity occurs in a room, it must for economical reasons be utilized for heating the room, i.e. it must be taken into consideration in the ventilation system control. This is the case with the room temperature control system as shown in Fig. 1-2.

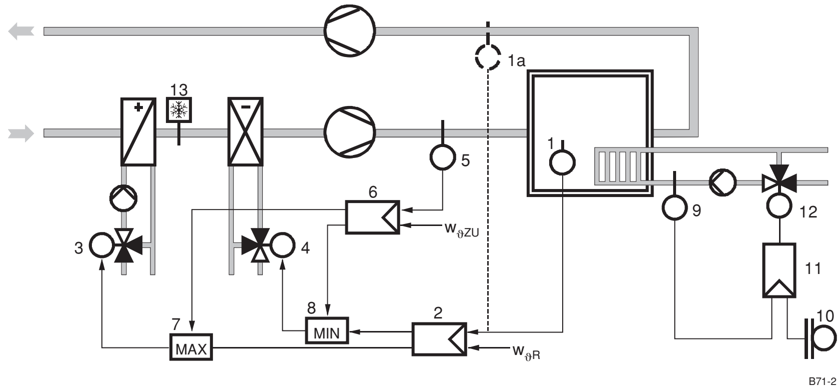


Fig. 1-2 Room temperature control with base load heating

1 Room temperature sensor (alternative: extract air temperature sensor 1a)
2 Room temperature controller
3 Motorized heating valve
4 Motorized heating valve
5 Supply air temperature sensor
6 Supply air temperature low limit controller
7 Heating valve priority selection
8 Cooling valve priority selection
9 Flow temperature sensor
10 Outdoor sensor
11 Flow temperature controller
12 Motorized heating valve
13 Frost protection thermostat

The control of this plant functions as follows:

- The supply air temperature is no longer controlled to a constant value but based on the room temperature, i.e. the modulating controller (2) compares the room temperature acquired by the sensor (1) with the setpoint. In case of a deviation, the controller adjusts the heating valve (3) or the cooling valve (4). The higher the room temperature tends to get because of internal heat sources, the colder the supply air that is blown in, so that the room temperature is kept constant at the selected value. A supply air temperature low limit controller (5) ensures that, in case of large internal heat gains, the supply air cannot become so cold that uncomfortable drafts occur)
- In order to prevent the system from expelling the internal heat unused along with the spent air, the base load heating must be reduced. This accommodates for the internal heat sources. However, the heat output of the radiators is only reduced to such an extent that the warm air curtain in front of the windows is still just preserved even under unfavorable conditions. For this purpose, the radiator heating system is generally designed for room temperatures of approximately 12...15 °C. However, it is important that the radiator heating system is controlled with compensation for ambient conditions
- The reduction in the heat output of the radiators means that the room can no longer by sufficiently heated in the absence of the internal heat sources. The heat output deficiency must, therefore, be compensated by the heating coil.

- Therefore, the heat output is divided up as follows:
  Heat output of the radiator heating system (base load heating)
  + Heat output of the ventilation system
  + Internal heat sources
  = 100 % heat output

Room temperature control of ventilation systems is used for heating and ventilating offices, conference, sales premises and store rooms as well as workshops and similar spaces with or without internal heat sources. The heating load can be divided between the base load heating and the ventilation system at any ratio (air curtain!). However, the base load heating system must not also be controlled based on the room temperature, otherwise the two room temperature control loops would interfere with each other.
The advantage of room temperature control over supply air temperature control lies mainly in the recovery time from disturbances occurring in the ventilated space:

- In the case of supply air temperature control, a disturbance occurring in the room is detected by the temperature sensor of the radiator heating system. Since this kind of heating generally reacts slowly, it takes a relatively long time for the disturbance to be cancelled out
- On the other hand, in the case of room temperature control, the disturbance is detected by the temperature sensor of the ventilation system. Since the room air is exchanged several times per hour (air change coefficient), the disturbance is cancelled out relatively quickly in comparison to the radiator heating system. However, disturbances that enter the room via the supply air take a relatively long time to cancel out because of the room's time constant. Among other things, this kind of disturbance can be:
  – a rapid change in the outside temperature
  – fluctuations in the flow temperature of the heat exchanger
  – the air washer switching on and off
  – changes in the position of air dampers that are controlled by a hygrostat

1.4 Extract air temperature control

Extract air temperature control has basically the same task as room temperature control, although the temperature sensor (1a) is installed in the extract air duct (shown with broken lines in Fig. 1-2.

In this regard, the following points must be noted, among others:
The temperature sensor installed in the extract air duct acquires the air temperature only, whereas a room temperature sensor also acquires a certain proportion of radiant heat, e.g. detected via the wall temperature, alongside the room air temperature. This fulfills the demands regarding comfort to a greater degree. On the other hand, the compensation time (T_g) of the controlled system is shorter with extract air temperature control, because the sensor reacts more quickly at higher flow rates. The dead time T_u of the controlled system does not become shorter, however, so the degree of difficulty S of the controlled system increases ($S = T_u/T_g$).

Extract air temperature control is preferred in applications where the proper placement of a room temperature sensor is problematical or where only the room air temperature, without any contribution made by radiant heat, is significant (e.g. in laboratories, operating theatres etc.).

1.5 Room/supply air temperature cascade control

As described in section 1.3, in the case of room temperature control, disturbances introduced by the supply air can only be detected and compensated by the room temperature sensor. This disadvantage is eradicated by cascade control. It also improves the controllability of difficult controlled systems.

1.5.1 Makeup

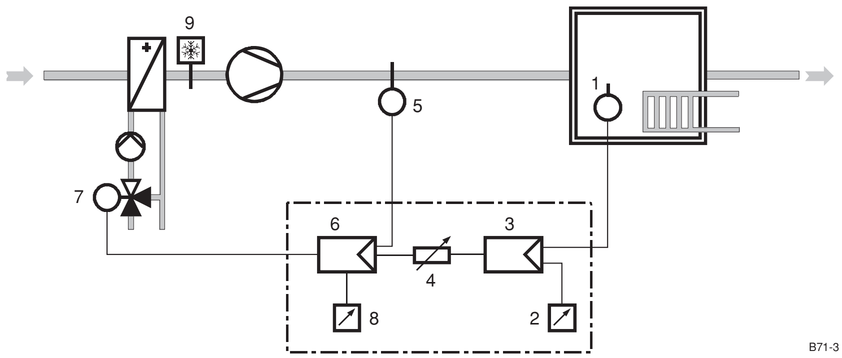


Fig. 1-3 Room/supply air temperature cascade control

1	Room temperature sensor
2	Room temperature setpoint adjuster
3	Master controller
4	Cascade influence adjusting potentiometer
5	Supply air temperature sensor
6	Slave controller (supply air temperature controller)
7	Motorized heating valve
8	Cascade base value adjusting potentiometer
9	Frost protection thermostat
1-2-3-4	Master control loop
4-5-6-7	Auxiliary or slave control loop

Master control loop

The master control loop consists of the master controller (3) (room temperature controller), room temperature sensor (1), room temperature setpoint adjuster (2), supply air temperature setpoint (4) as the controller's output variable and the room as the controlled system. Therefore, the supply air temperature serves as the manipulated variable y of the master controller here. Instead of positioning an actuator, the supply air temperature setpoint is varied in order to initiate the correction of the room temperature deviation. The slave control loop is made up of the supply air temperature setpoint (4,8), supply air temperature sensor (5), supply air temperature controller (6), heating valve (7) as the actuating device and the controlled system between the heating valve and the supply air temperature sensor.

Slave control loop

If a P-controller is used as the master controller and a PI-controller as the slave controller, the control system is referred to as P+PI cascade control. The following description covers only the behavior of this P+PI cascade control.

1.5.2 Mode of operation

- The master controller (3) acquires the room temperature via the sensor (1). Provided no deviation from the setpoint (2) occurs, the slave controller (6) (PI controller) keeps the supply air temperature constant at the setpoint specified by the room temperature controller (3) using the supply air temperature sensor (5). Supply air temperature disturbances are detected by the supply air temperature sensor and corrected by the slave controller, independent of the load, before they can affect the room temperature. The fact that supply air temperature disturbances are prevented from affecting the room temperature makes the task of the master controller easier. This controller now only needs to compensate for disturbances that occur in the room itself
- In case of a deviation of the room temperature (1) from the setpoint (2), the master controller (3) (P controller) shifts the supply air temperature setpoint in the opposite direction in proportion to the control deviation and the set cascade influence (4). With a cascade influence of 20 %, for example, the supply air temperature variation must be five times greater than the room temperature deviation that occurred, i.e. if the room temperature falls 1 K below the setpoint (2), the supply air temperature setpoint is raised 5 K over the set value (8, see Fig. 1-4).
- The slave controller (6) acquires the supply air temperature via the sensor (5), compares it with the new setpoint, and adjusts the valve (7) until the required supply air temperature is reached again
- Being a P-controller, the master controller cannot usually fully correct a room temperature disturbance; a proportional offset remains. In the case of P+PI cascade control, however, the proportional offset is considerably smaller than with P-control only, because the supply air temperature setpoint is maintained precisely by the PI slave controller. Additionally, since the transfer coefficient K_S of room controlled systems is usually < 0.5, the gain of the room temperature controller can be set relatively high. In practice, this means a maximum proportional offset of < 1 K

In summary, the following can be said about P+PI cascade control:
The master controller varies the supply temperature setpoint in proportion to the room temperature deviation. This gives rise to a proportional offset of the room temperature. However, the supply air temperature is controlled independent of the load by the PI slave controller. Disturbances in the supply air controlled system are corrected before they can affect the room temperature. In HVAC engineering, the master and slave controllers are combined in a single control device or software function block (chain-dotted box in Fig. 1-3).

1.5.3 Setting possibilities

- Setpoint (2) for the desired room temperature (w_R)
- Setting value (8) for the supply air temperature (cascade base value w_K): this is set to the value that is required to maintain the desired room temperature under normal conditions. It is independent of ventilation system design, base load heating of the space, internal heat gains in the space and also ambient conditions, provided the base load heating control does not have compensation for ambient conditions
- If base load heating is present, and the controlled system is only used for room ventilation, the cascade base value w_K can be set to approximately the same value as the room temperature setpoint w_R

- If the controlled ventilation system is also used for room heating (room without base load heating), the cascade base value w_K should be set higher than the room temperature setpoint w_R (approx. half load)
- In case of a continuous internal heat gains, the cascade base value w_K should be set lower than the room temperature setpoint w_R
- Cascade authority (4, E %): This value depends on ventilation system design. The smaller the authority is set, the greater the gain of the room temperature controller will be, i.e. the greater the amount by which the supply air temperature must change in order to correct a room temperature control deviation. Room temperature disturbances are corrected quickly, and the proportional offset becomes correspondingly smaller. The higher the authority is set, the more stable the room temperature will be, but the proportional offset will also be correspondingly larger. Generally, empirical values set in case where the conditions are unknown are around 15 %.

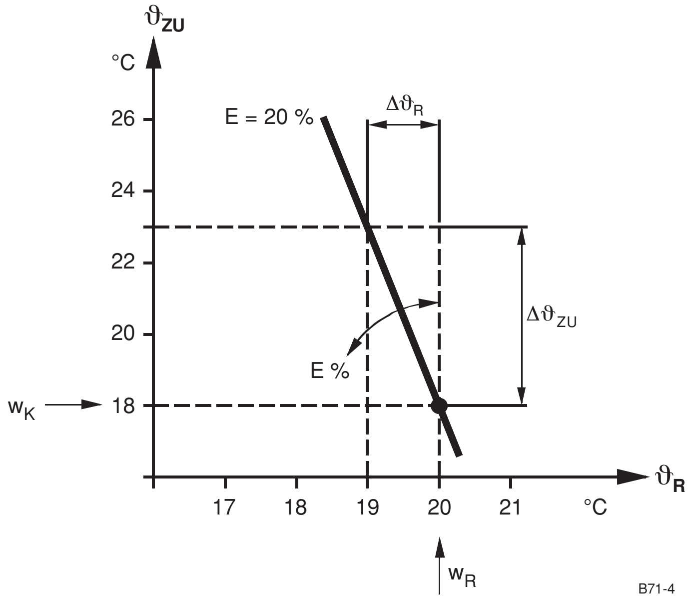


Fig. 1-4 Cascade authority

ϑ_R	Room temperature
w_R	Room temperature setpoint
$\Delta\vartheta_R$	Room temperature deviation
ϑ_{ZU}	Supply air temperature
w_K	Cascade base value (supply air temperature setting value)
$\Delta\vartheta_{ZU}$	Supply air temperature change
E %	Cascade authority in %

Controller gain K_P = cascade slope $S = \frac{\Delta\vartheta_{ZU}}{\Delta\vartheta_R}$

Cascade authority $E[\%] = \frac{100\ \%}{S} = \frac{\Delta\vartheta_R}{\Delta\vartheta_{ZU}} \cdot 100\ \%$

PI-PI cascade control

Most software programs today normally have PI-PI cascade control implemented. The same basic considerations apply to these systems as to the P-PI cascade control described above.

1.6 Sequence control of heating valve and cooling valve

The system shown in Fig. 1-5 is equipped with an air heating coil and an air cooling coil. This allows the room temperature to be maintained at the desired value (w_R) not only in winter but also in summer and in case of internal heat gain.

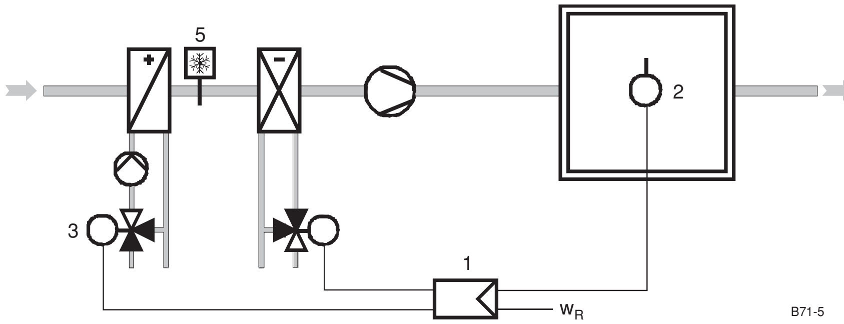


Fig. 1-5 Ventilation system with air heating coil and air cooling coil
1 Temperature controller
2 Room temperature sensor
3 Motorized heating valve
4 Motorized heating valve
5 Frost protection thermostat

The temperature controller (1) compares the room temperature acquired by the sensor (2) against the setpoint wR. In case of a deviation (Fig. 1-6a), the controller adjusts the heating valve (3) or the cooling valve (4) until the room temperature setpoint is reached. Therefore, the room temperature is controlled to the same setpoint wR both in heating and in cooling operation.

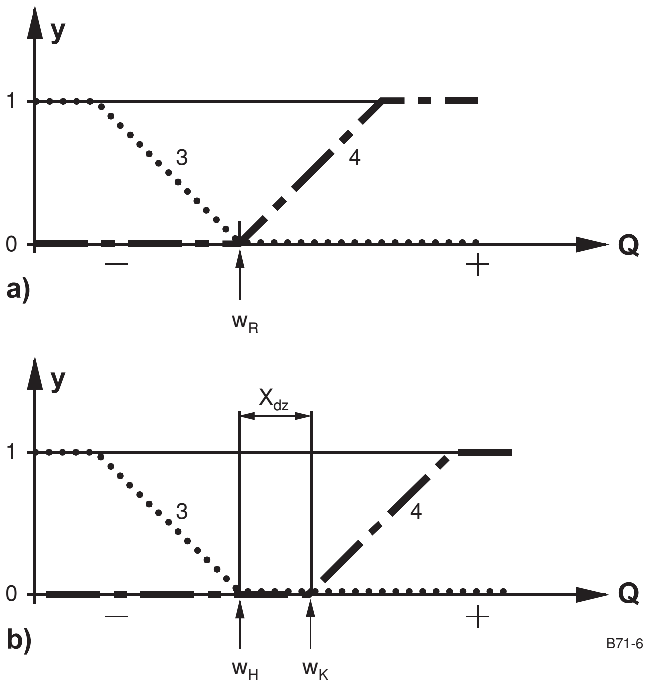


Fig. 1-6 Heating valve – cooling valve sequential control
a) Heating – cooling without dead zone
b) Heating – cooling with dead zone
Q Load (– = heating load, + = cooling load)
3 Heating valve
4 Cooling valve
w_R Room temperature setpoint
w_H Heating setpoint
w_K Cooling setpoint
x_{dz} Dead zone

It is, however, also possible to allow the room temperature to rise during cooling operation approximately 4 K above the heating setpoint w_H before the cooling valve starts to open (Fig. 1-6 b). The purpose of separating cooling and heating operation with the so-called dead zone x_{dz} is to save cooling energy. This means that the controller operates to the lower heating setpoint w_H during heating operation and to the higher cooling setpoint w_K during cooling operation.

It is recommendable to use a controller with PI action for this dead zone control, because then the proportional offset of the P-controller does not have to be taken into consideration, allowing the dead zone between the two setpoints w_H and w_K to be fully utilized, i.e. without energy consumption.

1.7 Temperature control with outside temperature compensation

In ventilation and air conditioning systems, room, extract air, supply air or cascade control setpoints can be governed by the outside temperature. This means that the temperature control setpoint is continuously varied within a specific outside temperature range.

Outside temperature compensation of a room temperature setpoint is implemented for the following reasons:

- During summer operation, the respective temperature setpoint is continuously raised as the outside temperature rises from say 20 °C to 32 °C (summer compensation) in order to avoid too great a difference between the room temperature and the outside temperature and, therefore, to prevent the risk of a heat shock. Additionally, cooling energy use is also reduced
- According to the VDI directives for ventilation, the room temperature should be raised to a maximum of 26 °C when the outside temperature is 32 °C. The following room temperature setpoints apply as per VDI for the outside temperature range from 20 °C to 32 °C:

Outside air temp.	°C	20	22	24	26	28	30	32
Room temperature	°C	20	21	22	23	24	25	26

- During winter operation, when the outside temperature is below 0 °C, for example, the room temperature control setpoint is also continuously raised (winter compensation) in order to compensate for the effects of lower surface temperatures of walls and windows on comfort (heat radiation) and for the proportional offset with P-controls
- Fig. 1-7 shows a schematic of a room temperature control system with outside temperature compensation. The outside temperature acquired by the temperature sensor (3) gives rise to a setpoint increase in the temperature controller (1) in summer and/or in winter. The starting point for summer compensation and winter compensation as well as the magnitude of the summer compensation authority and the winter compensation authority can be set individually on the controller (1). The setpoint change is directly proportional to the outside temperature during summer operation, and inversely proportional to it during winter operation

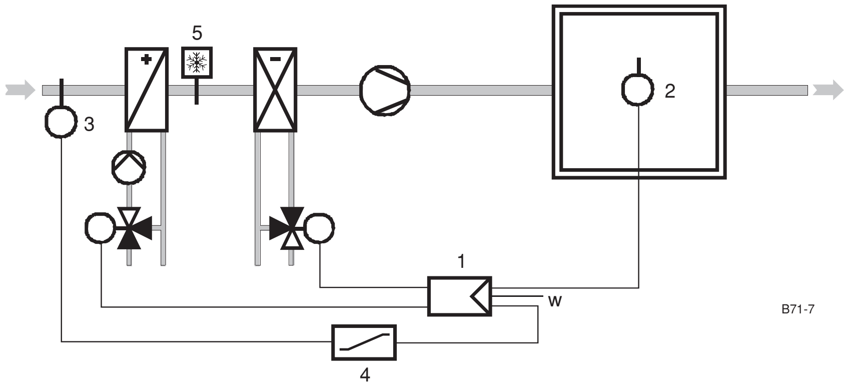


Fig. 1-7 Room temperature control with outside temperature compensation
1 Temperature controller with outside temperature compensation
2 Room temperature sensor
3 Sensor
4 Setpoint shift controller
5 Frost protection thermostat

Fig. 1-8 shows the schematic of a room temperature control system with outside temperature compensation. The following are normal setting values for the outside temperature shift controller with modulating controllers:

Parameter:	**P-controller:**	**PI/PID-controller:**
Summer compensation ($E_2 = \Delta\vartheta_R/\Delta\vartheta_{AU} * 100$ %)	40 %	50 %
Winter compensation ($E_1 = \Delta\vartheta_R/\Delta\vartheta_{AU} * 100$ %)	12–15 %	5 %
Summer compensation starting point (w_{FS})	Room temperature setpoint	
Winter compensation starting point (w_{FW})	0...5 °C	

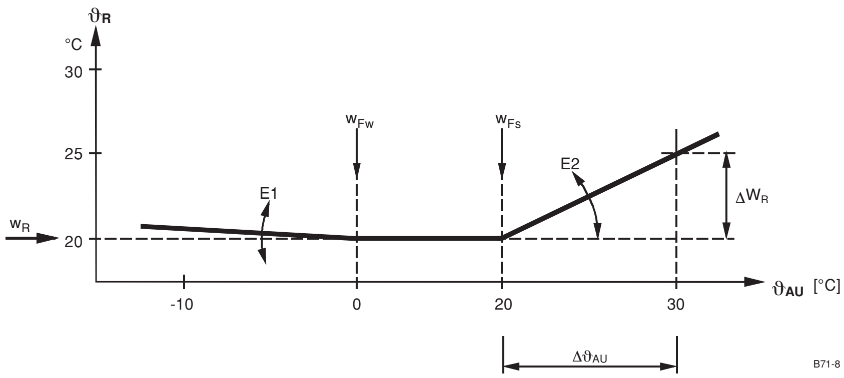


Fig. 1-8 Authority of outside temperature compensation
ϑ_R Room temperature
Δw_R Room temperature setpoint change
ϑ_{AU} Outside temperature
$\Delta\vartheta_{AU}$ Outside temperature change
w_R Room temperature setpoint
w_{FS} Summer compensation starting point
w_{FW} Winter compensation starting point
E_1 Winter compensation authority in %
E_2 Summer compensation authority in %

The summer or winter compensation authority is calculated according to the following formula:

$$E = \frac{\Delta\vartheta_R * 100}{\Delta\vartheta_{AU}} \; [\%]$$

1.8 Minimum limitation of the supply air temperature

In ventilation and air conditioning systems with room temperature control, a temperature controller with a sensor in the supply air duct is used to limit the supply air temperature to a minimum value (Fig. 1-9). This is necessary where the controller tends to reduce the supply air temperature sharply in order to maintain the room temperature setpoint in case of large internal heat gains. Without this limitation, the supply air temperature would reach such low values that unhealthy drafts would occur.

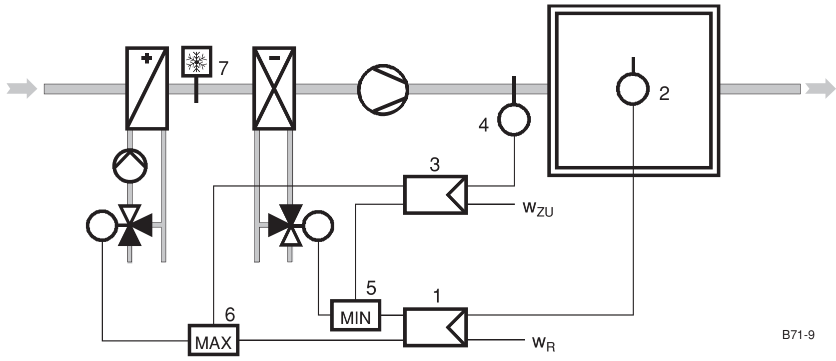


Fig. 1-9 Low limit room temperature control with minimum limitation of the supply air temperature

1 Room temperature controller
2 Room temperature sensor
3 Low limit controller
4 Supply air temperature controller
5 Heating valve priority selection
6 Cooling valve priority selection
7 Frost protection thermostat

The low limit control of the supply air temperature is accomplished as follows:
The duct temperature sensor (4) is installed in the supply air duct near the room outlet and connected to the limit controller (3). The limit value is set to 17...18 °C in the controller. It must be low enough to dissipate the internal heat, but not so low that drafts occur. If the supply air temperature falls below the set limit value, the limit controller takes over the closing of the cooling valve via the <MIN> priority selection and the opening of the heating valve via the <MAX> priority selection, preventing any further drop in the supply air temperature. When the limit value is reached, this control concept switches over from room temperature control to supply air temperature control, which then keeps the supply air temperature constant at the limit value (Fig. 1-10).

The control gain K_P (or proportional band X_P) of the room temperature controller must be adapted to the time lag of the room controlled system, and that of the limit control must be adapted to the degree of difficulty of the fast duct controlled system.

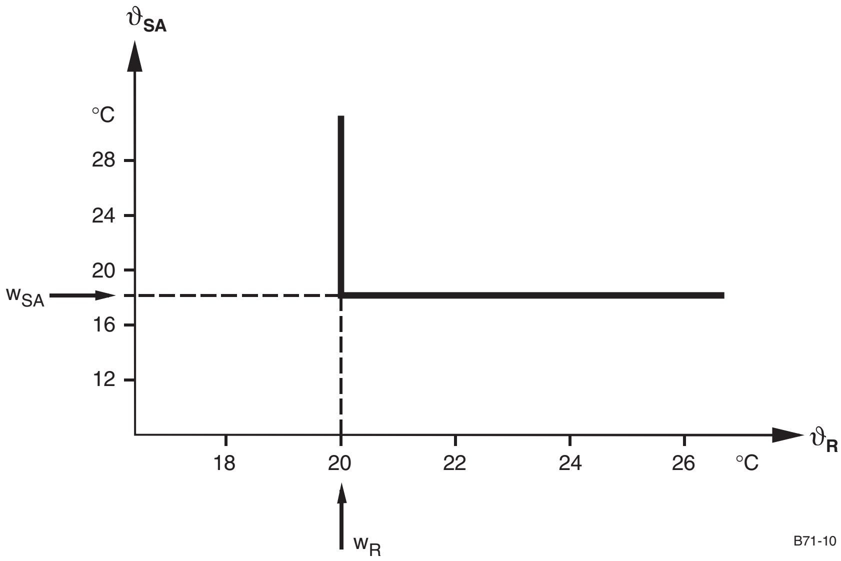


Fig. 1-10 Minimum limitation of the supply air temperature

ϑ_R Room temperature

ϑ_{SA} Supply air temperature

w_R Room temperature setpoint

w_{SA} Supply air temperature limit values

2. Humidity controls

2.1 General

The task of an air conditioning system is to keep the temperature and relative humidity of the air in the conditioned space constant at the specified values or within specified limits, regardless of the state of the outside air and the processes within the space. Therefore, appropriate control systems are used for the heating, cooling, humidification and dehumidification of the air. In the case of humidity control, a distinction is mainly made between:

- indirect humidity control via the dew point temperature of the desired room air state or via the water vapor content x of the supply air
- direct humidity control with a relative humidity sensor in the conditioned space

2.2 Dew point control

The prerequisite for dew point control is an air washer that has a humidification efficiency of at least 95 %, i.e. one that can almost achieve the saturation of the outflowing air. If the temperature of this practically saturated air is also controlled, its water vapor content is fixed. Therefore, the required effort in terms of control equipment is relatively small. This solution is appropriate in applications where the air that has been cooled to the dew point is largely reheated by internal heat sources in the conditioned space (e.g. heat from machines in industrial facilities). If this is not the case, direct humidity control is, for economical reasons (energy costs), preferable in conventional air conditioning systems.

2.2.1 Makeup

Fig. 2-1 shows the makeup of a dew point control system:

- The dew point control system works with the temperature controller (2) and the temperature sensor (1) directly downstream from the air washer. The controller's output signals operate the preheater valve and the cooling valve. The actuators of the two valves are controlled sequentially
- The circulated water volume in the humidifier remains constant, and the humidification pump is constantly on during system operation

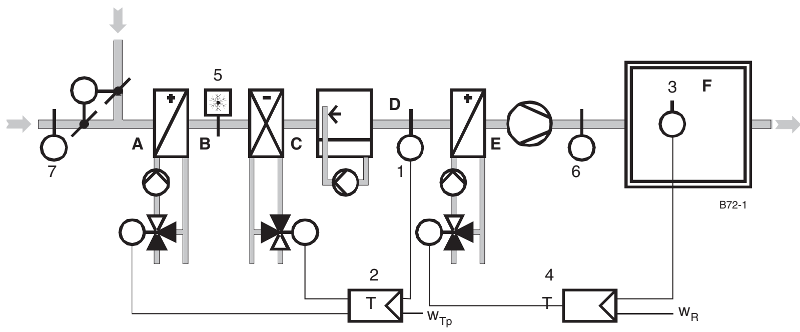


Fig. 2-1 Dew point control

1 Dew point temperature sensor
2 Dew point temperature controller
3 Room temperature sensor
4 Room temperature controller
5 Frost protection thermostat
6 Sensor for supply air temperature low limit controller
7 Sensor for outside temperature shift controller
A...F Air states on the psychrometric chart

- In conventional systems, the control of the temperature and, therefore, indirectly of the room relative humidity by dew point control requires a reheater whose heat output is controlled by the room temperature controller
- The system can be complemented with a supply air temperature low limit controller and outside temperature shift controller as well as outdoor/recirculated air dampers

2.2.2 Control

Using the dew point temperature sensor (1) downstream from the humidifier, the dew point temperature controller (2) ensures that the dew point temperature setpoint is maintained with the various air states. In case of deviation of the dew point temperature from the setpoint w_{TP}, the controller actuates the preheater valve or cooling valve. In order to achieve the desired temperature and relative humidity in the conditioned space, a second control loop with controller (4) and room temperature sensor (3) is required to control the humidified and cooled air to the desired inlet temperature via the reheater valve.

If the precise maintenance of a specific relative humidity has priority over the room temperature, e.g. in industrial systems (chemical, textile industry etc.), a room humidity sensor can be used in the reheater control loop instead of the room temperature sensor.

2.2.3 Application Dew point control is only used in systems where the room relative humidity is permitted to fluctuate within certain limits, e.g. in offices, conference rooms, showrooms etc., because variations in the internal heat gains, room temperature setpoint changes and humidity disturbances (external humidity sources, humidity loss) are only detected by the control system if auxiliary control loops are implemented. Dew point control is mainly used in industrial applications for adiabatic cooling of waste heat from machines, and in comfort applications for central air treatment in dual duct, induction unit or fan coil systems (Fig. 2-2).

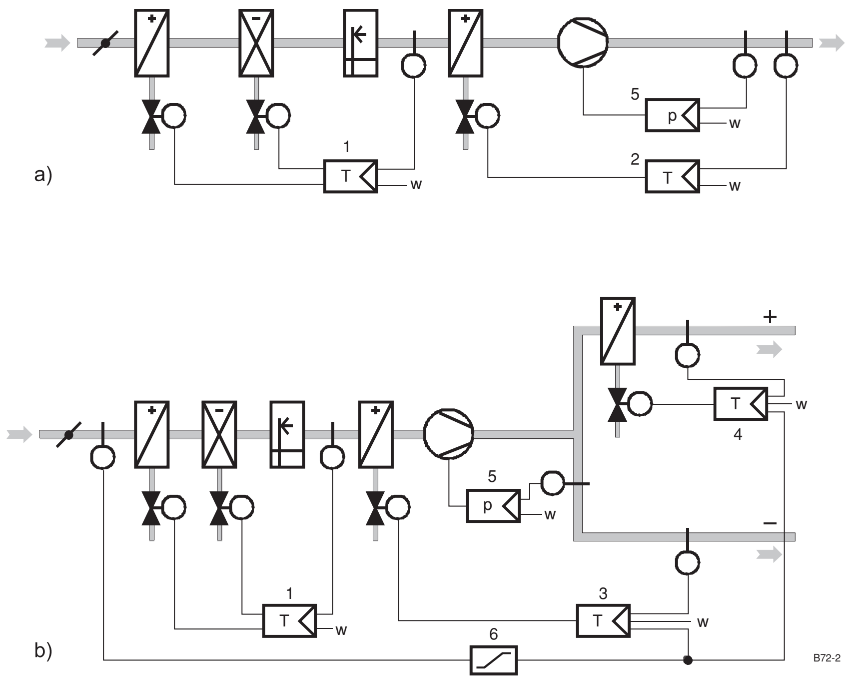


Fig. 2-3 Central air treatment systems with dew point control
- a) Central primary air treatment (e.g. for induction units)
- b) Central air treatment for a dual-duct system
- 1 Dew point control
- 2 Supply air temperature control in the primary air system
- 3 Cool air temperature control in the dual-duct system
- 4 Warm air duct control in the dual duct system
- 5 Supply air duct pressure control
- \+ Warm air duct
- – Cool air duct

2.2.4 Dew point temperature

The desired temperature (1, e.g. 20 °C) and relative humidity (2, e.g. 50 %) in the conditioned space give rise to the dew point temperature ϑ_{DP} (3) of about 9 °C on the psychrometric chart (Fig. 2-3). The dew point temperature serves as the setpoint of the dew point temperature controller, regardless of the efficiency of the humidifier.

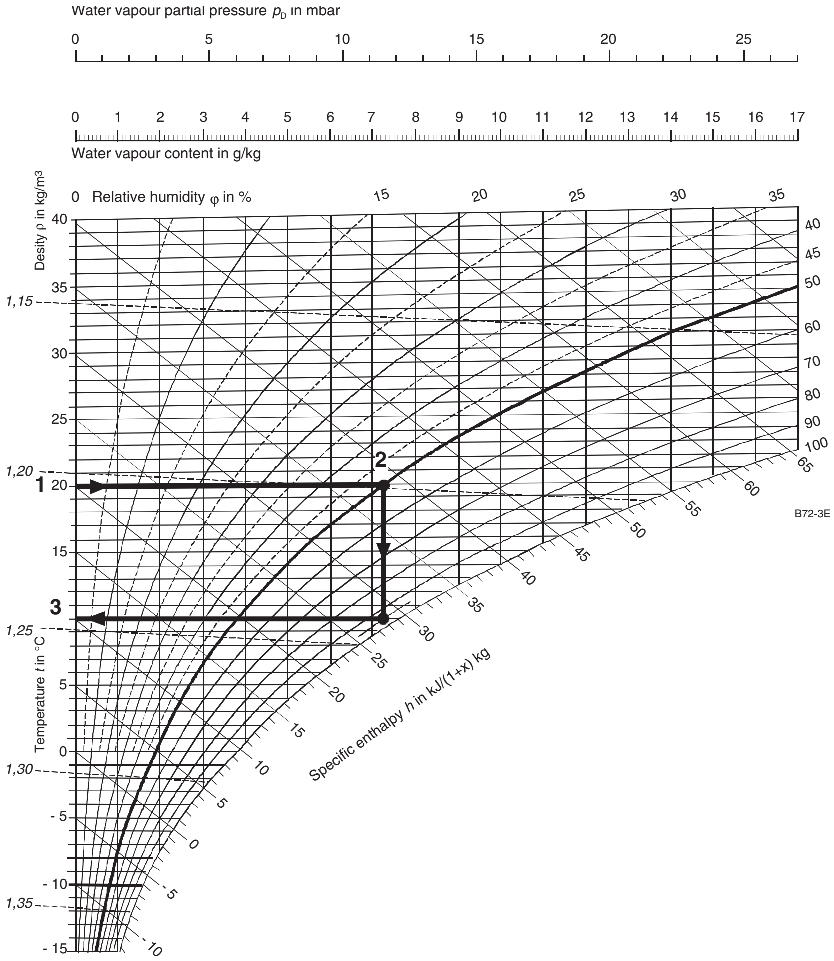


Fig. 2-3 Determining the dew point temperature ϑ_{DP} on the psychrometric chart
1 Desired room temperature
2 Desired room air humidity
3 Resulting dew point temperature ϑ_{DP}

2.2.5 Air state changes on the psychrometric chart

On the system schematic (Fig. 2-1), air states of interest are identified with letters which also appear on the psychrometric chart (Fig. 2-4 and Fig. 2-5).

Air states in winter

(Fig. 2-7) The intake air A is heated to state B in the preheater. Then, the air is humidified almost to the saturation curve, i.e. to a relative humidity of approximately 95 %, in the humidifier. In the process, it cools to the desired dew point temperature D. Therefore, the air must always be heated to such an extent by the preheater that it reaches the dew point temperature after humidification (task of the dew point temperature control loop).

Finally, the air must be heated to such an extent in the reheater (state E) that it covers the room heat requirement, giving rise to the desired temperature and relative humidity of the conditioned space (state F, task of the room temperature control loop).

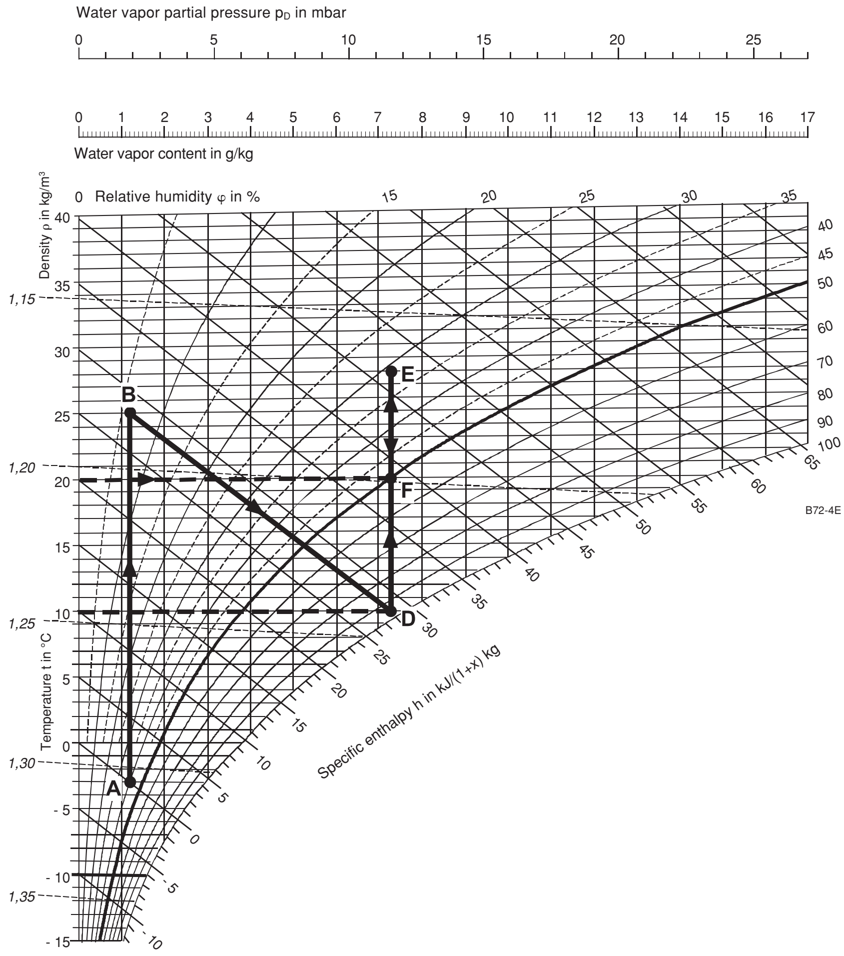


Fig. 2-4 State changes with dew point temperature control in winter
- A Preheater inlet state
- B Air washer inlet state
- D Supply air dew point
- E Supply air inlet state
- F Desired state of the room air

Air states in summer

(Fig. 2-5) The intake air A (A = mixing point after outside/recirculated air mixing) is cooled to state C in the cooling coil and, if the cooling surface temperature (2) is below the dew point temperature of air state A, it is also dehumidified. The air is then humidified to the saturation curve (φ ≈ 95 % r.h.) in the air washer. In the process, it is further cooled to the dew point temperature D. Therefore, the air must always be cooled to such an extent by the cooling coil that it reaches the dew point temperature downstream from the air washer (task of the dew point temperature control loop).
Finally, the air is reheated to such an extent in the reheater (point E) that it covers the room cooling load, giving rise to the desired temperature and relative humidity of the conditioned space (state F, task of the room temperature control loop).

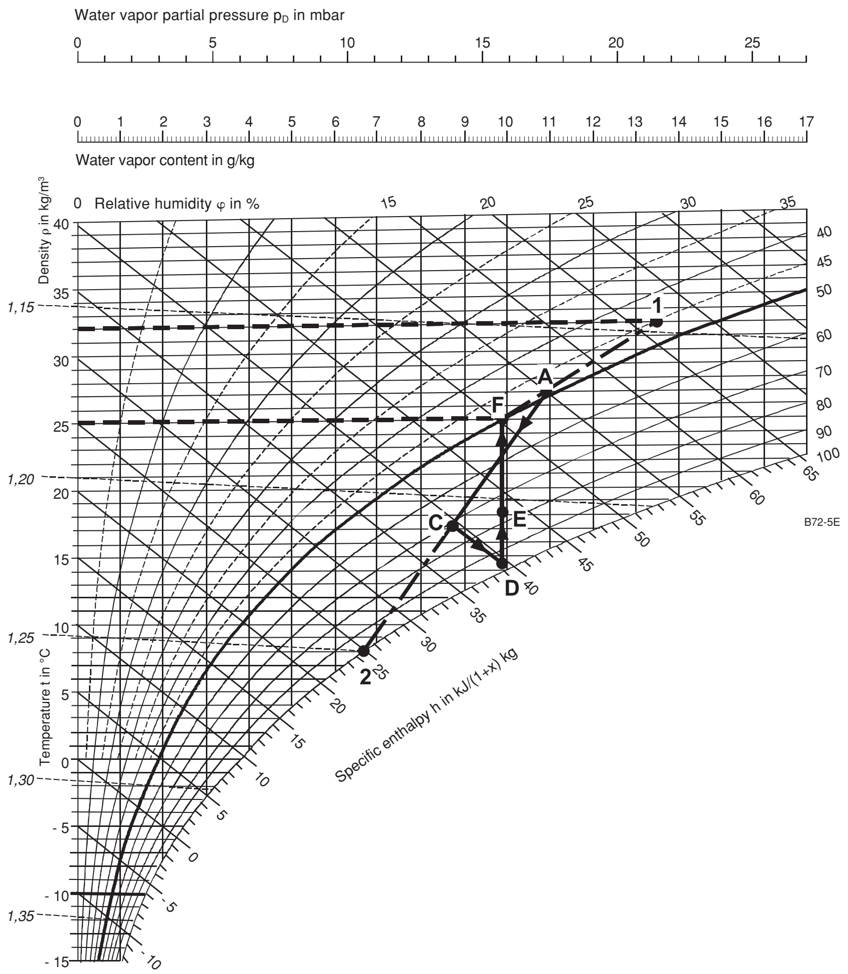


Fig. 2-5 State changes with dew point temperature control in summer
A Cooling coil inlet state
C Air washer inlet state
D Supply air dew point
E Supply air inlet state
F Desired state of the room air
1 Outside air state in summer
2 Mean cooling coil surface temperature

2.2.6 Properties of dew point control

- In case of a change of the room temperature due to a setpoint adjustment or due to the authority of the outside temperature shift controller, the room relative humidity changes. Additionally, humidity disturbances in the conditioned space (e.g. external humidity sources) are not detected. If, however, the room humidity level is to remain constant, the dew point temperature must be adjusted accordingly
- The room relative humidity also depends on the humidification efficiency of the air washer. What this means in practice is that, in case of a lower humidification efficiency, the setpoint of the dew point temperature controller must be set correspondingly higher than the theoretical dew point temperature of the desired room air state (Fig. 2-6)

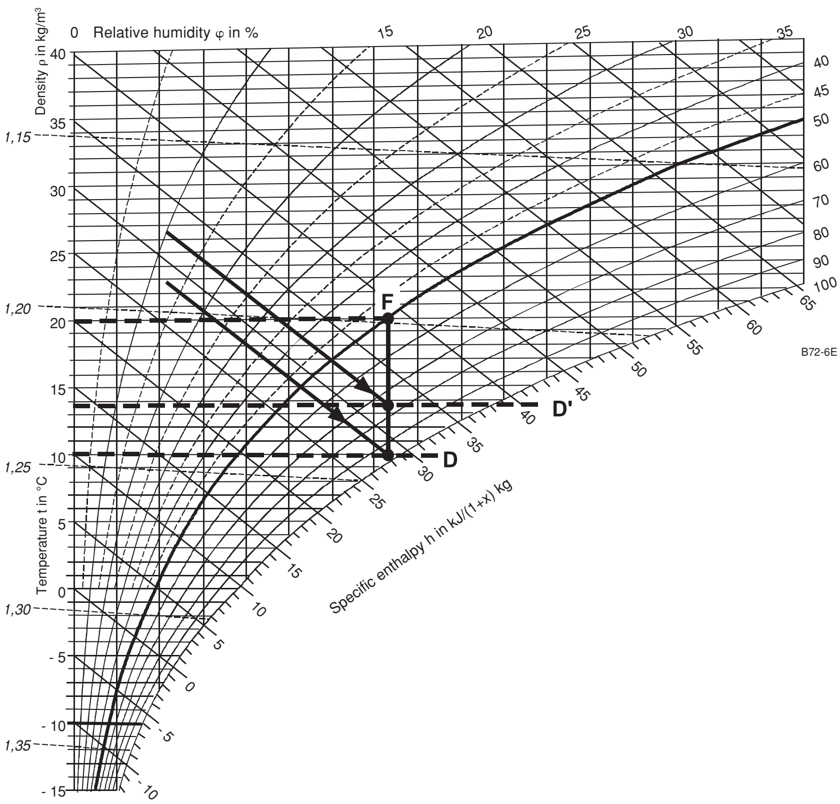


Fig. 2-6 Effect of the humidification efficiency on the dew point temperature setpoint
D Dew point temperature setpoint for a humidification efficiency of approx. 95 %
D' Dew point temperature setpoint for a humidification efficiency of approx. 75 %
F Desired room air state (20 °C / 50 % r.h.)

2.2.7 Shifting dew point control based on outside temperature

Application

This type of control is applicable in cases where the room relative humidity must not be affected by an adjustment of the room temperature setpoint by the outside temperature shift controller.

Makeup

(Fig. 2-7a) The difference to the system shown in Fig. 2-1 is that the outside temperature compensation of the room temperature is also used to vary the dew point temperature. The same outside temperature shift controller (3) can be used for this purpose, because the variation of the two temperatures is practically parallel (Fig. 2-7b).

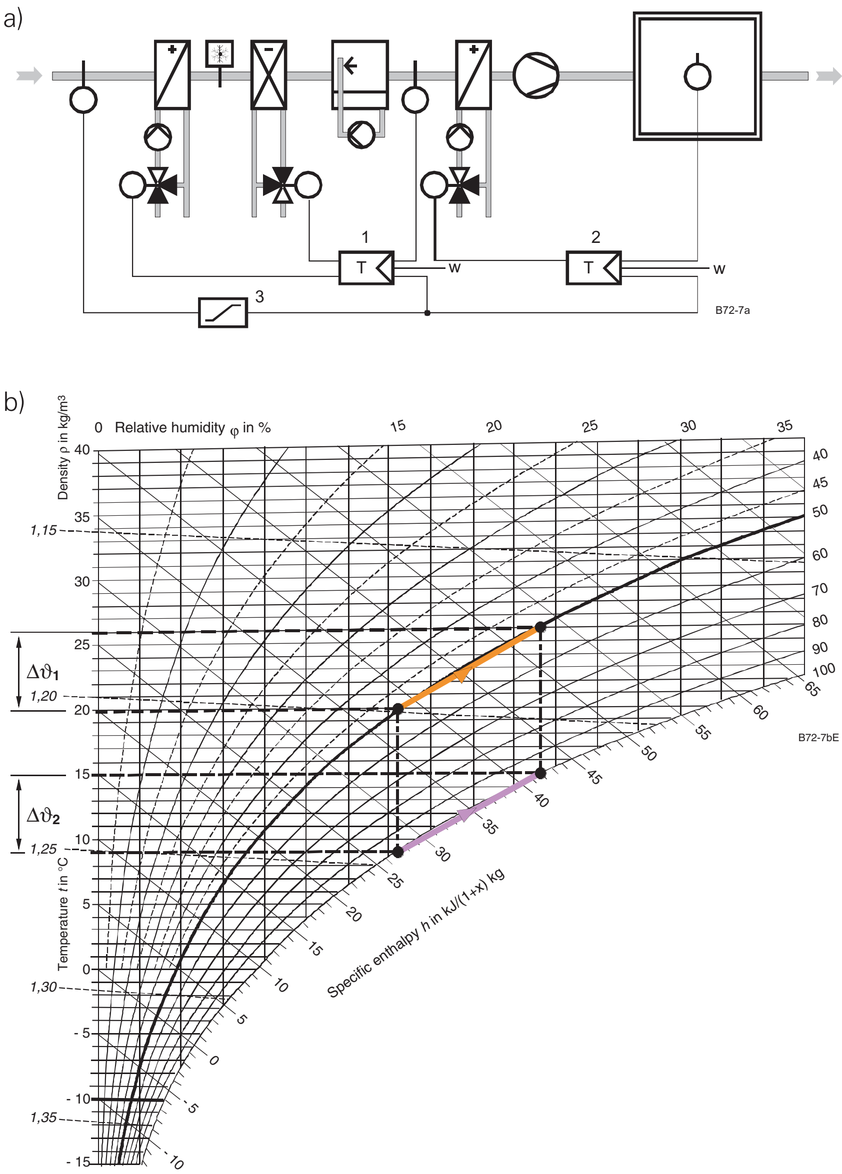


Fig. 2-7 Dew point temperature control with setpoint compensation based on outside temperature

a) Schematic
b) Setpoint compensation on the psychrometric chart
1 Dew point temperature controller
2 Room temperature controller
3 Outside temperature shift controller
$\Delta\vartheta_1$ Room temperature setpoint variation (20...26 °C)
$\Delta\vartheta_2$ Dew point temperature setpoint variation (9...15 °C)

- Disturbances of the room relative humidity due to the effects of the outside temperature shift controller no longer occur
- However, the room humidity can still be disturbed by:
  - External humidity sources
  - Room temperature changes via setpoint adjustment
  - Internal heat that can no longer be dissipated by the system (e.g. if the supply air temperature is being held at the low limit by the low limit controller)

2.3 Direct, modulating humidity control

2.3.1 General

Application

Direct humidity control is used in cases where the setpoints for the temperature and humidity of the conditioned space must be maintained exactly at all outside air states and in case of internal heat gains and external humidity.

Makeup

An air conditioning system is only able to exactly maintain the desired room temperature and relative humidity if a temperature control loop with a room temperature sensor and a humidity control loop with a humidity sensor in the conditioned space or in the exhaust air duct are used (Fig. 2-8). In this case, the actuating signals of the room temperature controller (2) act sequentially on the heating valve and cooling valve, and those of the humidity controller (4) act on the humidifier valve and – with priority control (5) for dehumidification purposes – also on the cooling valve.

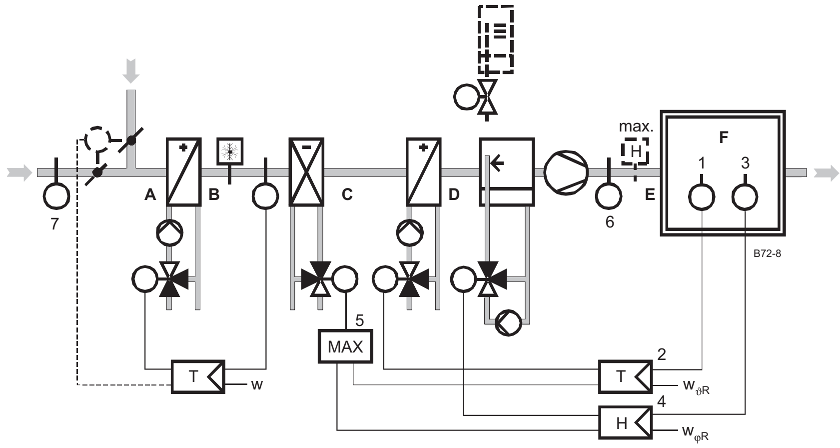


Fig. 2-8 Air conditioning system with direct humidity control

1	Room temperature sensor
2	Room temperature controller
3	Room humidity sensor
4	Humidity controller
5	Priority selector
6	Supply air temperature sensor for low limit control
7	Outside temperature sensor for shift controller
A..F	Air states

Humidification is generally accomplished using the following humidifier types:

- Controllable air washer
- Controllable steam humidifier

Both types provide for control to absolute humidity or relative humidity.

The air cooling coil has the dual function of cooling (under the control of temperature controller 2) and dehumidification (under the control of humidity controller 4). Therefore, it must be located upstream of the reheater and humidifier. Depending on operating state, it must be possible to provide cooling (temperature controller), humidification or dehumidification (humidity controller) and reheating (temperature controller). The sequence of the humidifier and reheater can also be reversed without affecting the proper functioning of the system.

A constant room temperature is maintained by the temperature controller (2) with the temperature sensor (1). The temperature controller controls the reheater valve and, via the priority selector (5), the cooling valve. A shift controller can be used to provide outside temperature (7) compensation of the temperature controller's setpoint. Additionally, a supply air temperature (6) low limit controller can be used.

The humidity control is accomplished by the humidity controller (4) using the humidity sensor (3) and the humidifying valve and cooling valve (dehumidification) as actuating devices. Since the cooling valve is controlled both by the temperature controller and the humidity controller, it must be automatically assigned using the priority selector (5) to the controller that is transmitting the largest positioning signal in each case, i.e. the one that is requesting the greatest cooling output.

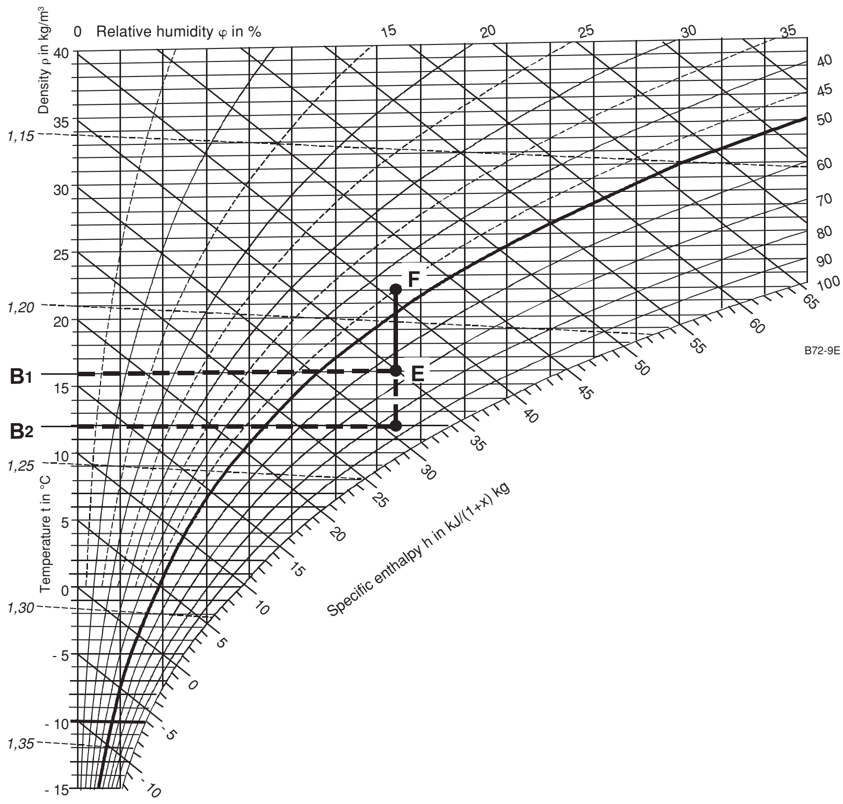
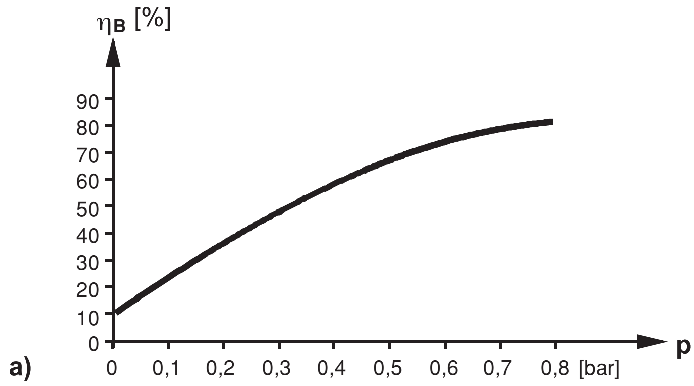


Fig. 2-9 Determining the setpoint for the preheater control loop
F Desired room air state
E Lowest permissible supply air temperature
B1 Maximum limit for the preheater control loop (lowest permissible supply air temperature)
B2 Minimum limit for the preheater control loop (to prevent condensation)

The separate preheater control loop provides frost protection for the cooling coil.
The air temperature downstream from the preheater must, on the one hand, be sufficiently low that the cooling valve never opens before the preheating valve is closed, and on the other, it must be sufficiently high that the required supply air temperature never falls below the dew point temperature when the reheater valve is closed. Therefore, the primary control setpoint in Fig. 2-9 is between points B1 and B2. A damper actuator for outside/recirculated air dampers can, of course, also be controlled in sequence with the preheater valve.

2.3.2 Air conditioning system with continuously controllable air washer



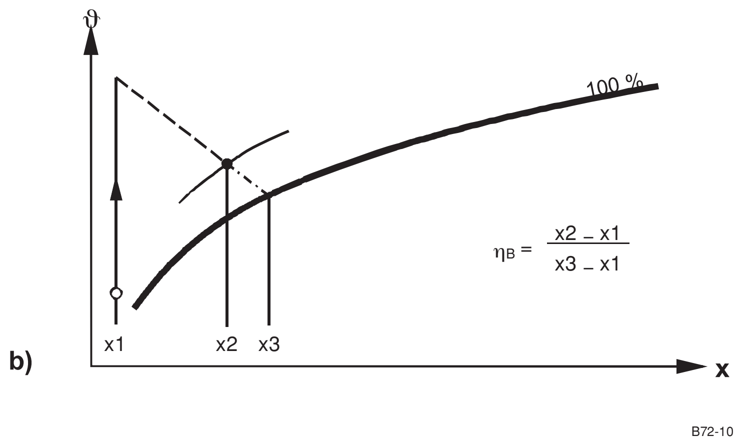


Fig. 2-10 Humidification efficiency of the air washer

a) Humidification efficiency as a function of nozzle pressure

b) Explanation of the humidification efficiency

ηB Humidification efficiency of the air washer

p Air washer nozzle pressure

x1 Assumed lowest water vapor content of the air

x2 Water vapor content of the air at room air state

x3 Water vapor content of the air at the saturation point

The humidity controller (4) continuously controls the humidification valve in proportion to the relative humidity (3) of the conditioned space (Fig. 2-8). This allows the humidification efficiency of the air washer to be infinitely reduced (Fig. 2-10), allowing the supply air to be humidified to the desired water vapor content. When the humidification valve is closed, the humidification pump is shut down via an auxiliary switch on the actuator.

On the system schematic (Fig. 2-8), air states of interest are identified with letters which are also used on the psychrometric charts (Fig. 2-11, Fig. 2-12 and Fig. 2-13) with the same meanings.

2.3.2.1 Air state changes in winter

Air heating (Fig. 2-11) The intake air A is heated to state B in the preheater. Since the cooling valve is closed, state B = state C. The air is further heated in the reheater until, after the degree of humidification Δx determined by the humidity controller (adiabatic cooling, but not to the saturation curve because controllable), the supply air has the necessary temperature (state E) in order to maintain the required room air state F. If the intake air temperature A is higher than state B, the preheater valve remains closed.

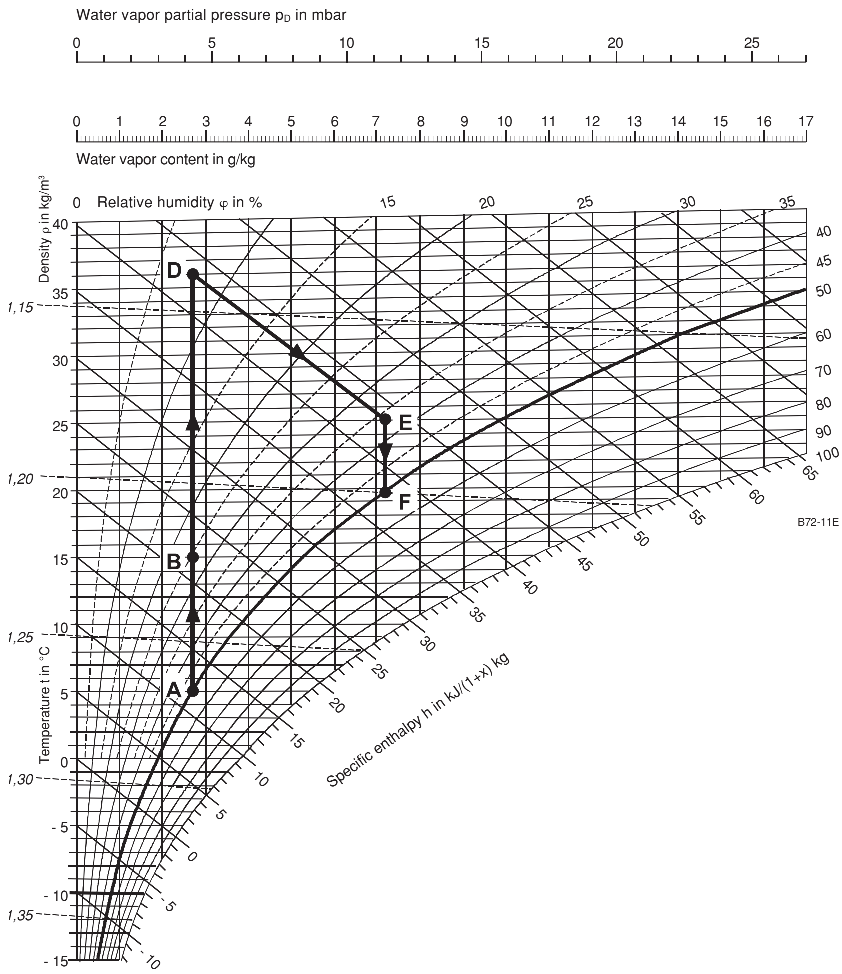


Fig. 2-11 State changes with direct humidity control (controllable air washer) in winter

- A Preheater inlet state
- B Reheater inlet state
- D Air washer inlet state
- E Supply air inlet state
- F Desired state of the room air

2.3.2.2 Air state changes in summer

Cooling The intake air A (state A = state B since the preheater valve is closed) is cooled in the cooling coil to the value commanded by the room temperature controller (state C = D since the reheater valve is closed).

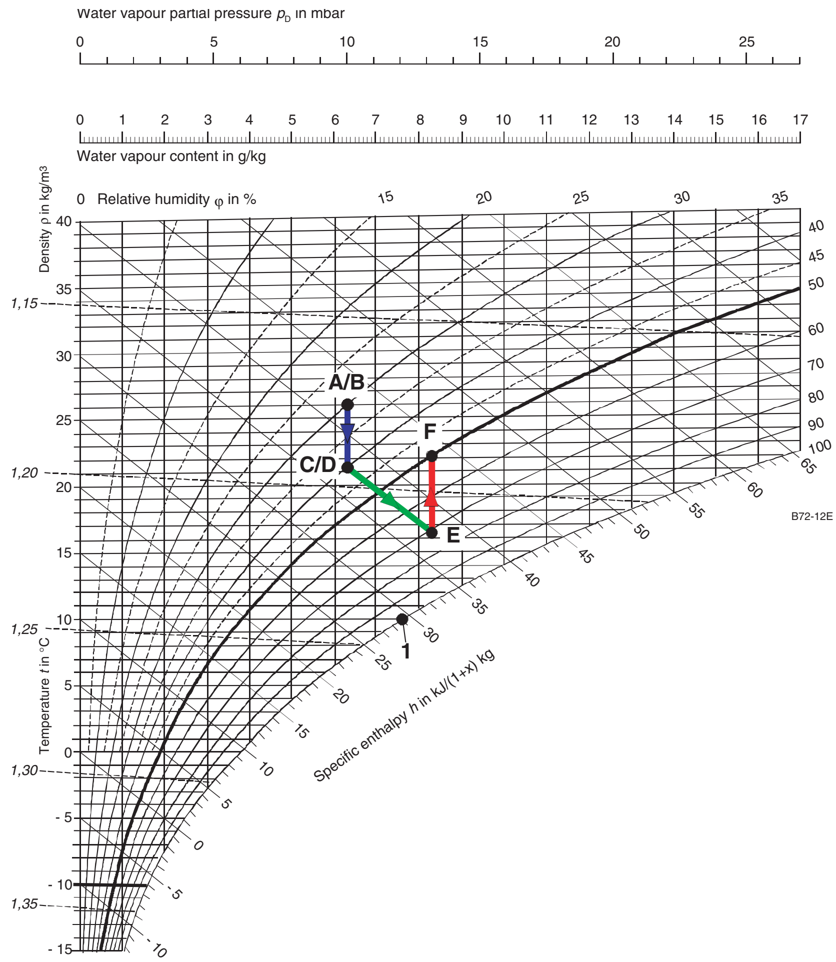


Fig. 2-12 State changes in case of cooling with direct humidity control

A	Preheater inlet state
B	Cooling coil inlet state
C	Reheater inlet state
D	Air washer inlet state
E	Supply air inlet state
F	Desired state of the room air
1	Mean cooling surface temperature ϑ_{CO}

Following the humidification as determined by the humidity controller (additional adiabatic cooling), this gives rise to the necessary inlet state E of the supply air in order to comply with the required room air state F. Since the mean cooling surface temperature ϑ_{CO} (1) is above the dew point of air state B in this case, cooling occurs without dehumidification.

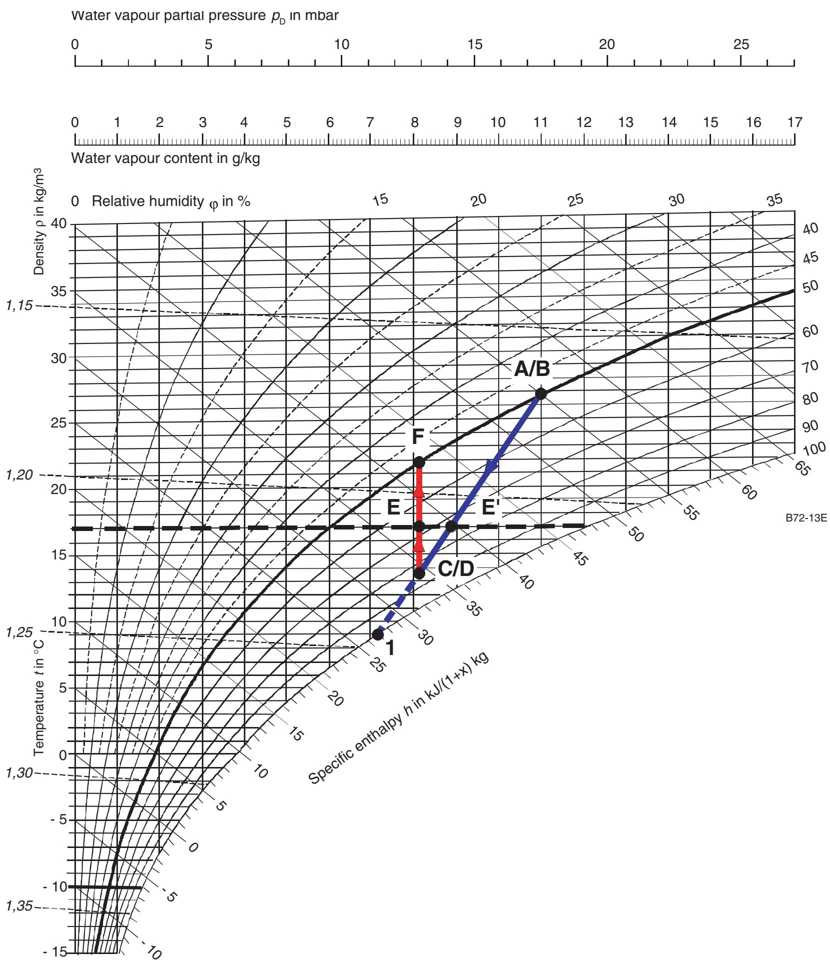


Fig. 2-13 State changes in case of cooling/dehumidification with direct humidity control

A	Preheater inlet state
B	Cooling coil inlet state
C	Reheater inlet state
D	Air washer inlet state
E′	Intersection of cooling characteristic with inlet temperature
E	Supply air required state
F	Desired state of the room air
1	Mean cooling surface temperature ϑ_{CO}

Cooling/dehumidification

Both the temperature and the absolute humidity of air state A (Fig. 2-13) are too high, which is why cooling with dehumidification is required in this case. In the initial phase, the temperature controller commands "cooling" and the humidity controller "de humidification." The controller with the greatest output signal determines the position of the cooling valve (via the priority function). The temperature controller commands cooling until the supply air temperature reaches the value required in order to maintain the desired room temperature (state E′). However, air state E′ is still too humid, which is why the humidity controller continues to command dehumidification (cooling). The continued dehumidification cools the supply air below the required inlet temperature. The humidity controller continues to command dehumidification until the water vapor content x necessary to maintain the required room relative humidity is achieved (state C/D). However, the room temperature controller must then heat the excessively cooled air up to the necessary supply air inlet temperature E via the reheater.

2.3.3 Continuously controllable steam humidification

In the case of steam humidification, the state change direction on the psychrometric chart corresponds to the heat content of the injected steam. The enthalpy increase (h of the air with the added quantity of steam Δx in grammes can be calculated in kJ using the steam table. In the case of humidification with saturated steam at approximately 100 °C (Fig. 2-14), the state change progression is approximately parallel to the isotherm. Therefore, only minor heating but very effective humidification of the air occurs. The other state changes occur in a similar manner to those with controllable water spray humidifiers.

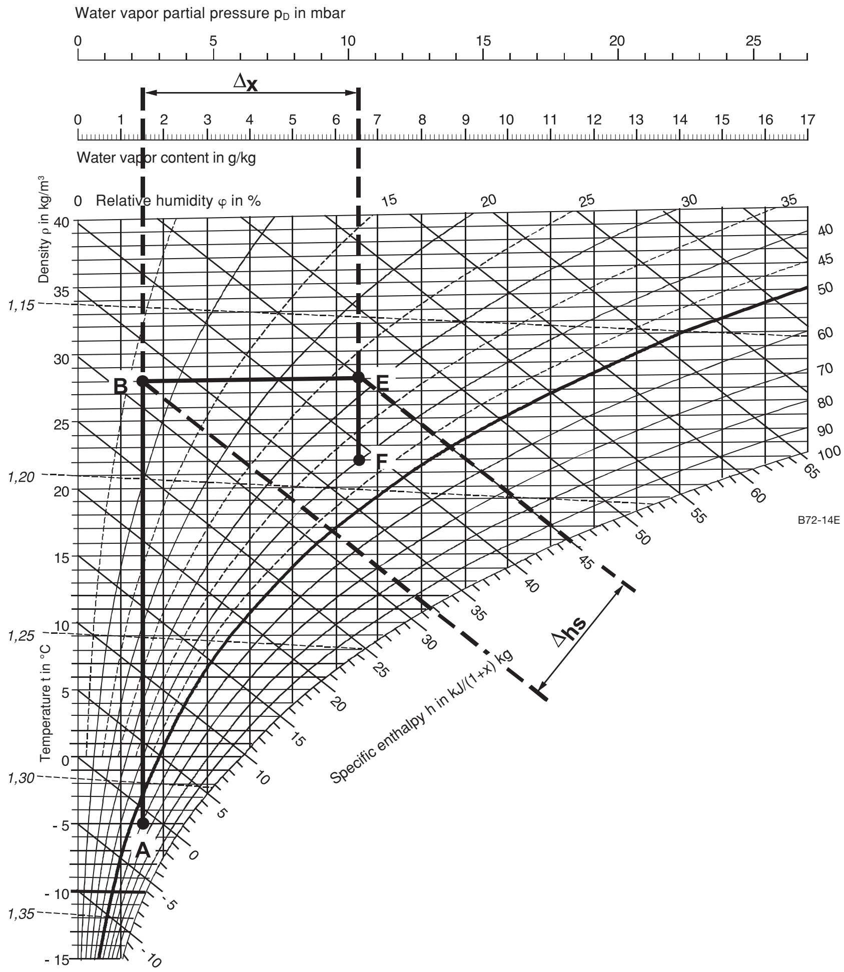


Fig. 2-14 State changes with continuously controllable steam humidification

A	Preheater inlet state
B	Steam humidifier inlet states (depending on enthalpy of the steam)
E	Supply air inlet state
F	Desired state of the room air
Δx	Specific steam supply in g/kg air
Δh_S	Increase in heat content by saturated steam at approx. 100 °C

2.3.4 Properties of direct, modulating humidity control

Direct, modulating humidity control has the following properties:

- The relative humidity (and temperature) of the conditioned space can be kept constant at any outside air state, with room temperature setpoint adjustments, internal heat sources, external humidity gain or humidity loss
- In the case of water spray humidification, a humidification efficiency that is considerably lower than that of the dew point air washer is sufficient. This means that a correspondingly lower priced device can be used. It is, however, important that the humidifier is controllable within a sufficiently large output range.
  The kind of unfavorable operating states that can occur with dew point control at partial loads are avoided. Only as much cooling, humidification and reheating as necessary occurs. This gives rise to lower operating costs.
  In this context, Fig. 2-15 shows 2 different air state changes for comparison between dew point control and direct humidity control with a controllable, adiabatic water spray humidifier.
  Dew point control with energy waste Δh due to unnecessary cooling, humidification and reheating.
  Humidity control with a controllable, adiabatic humidifier (only as much cooling as necessary)

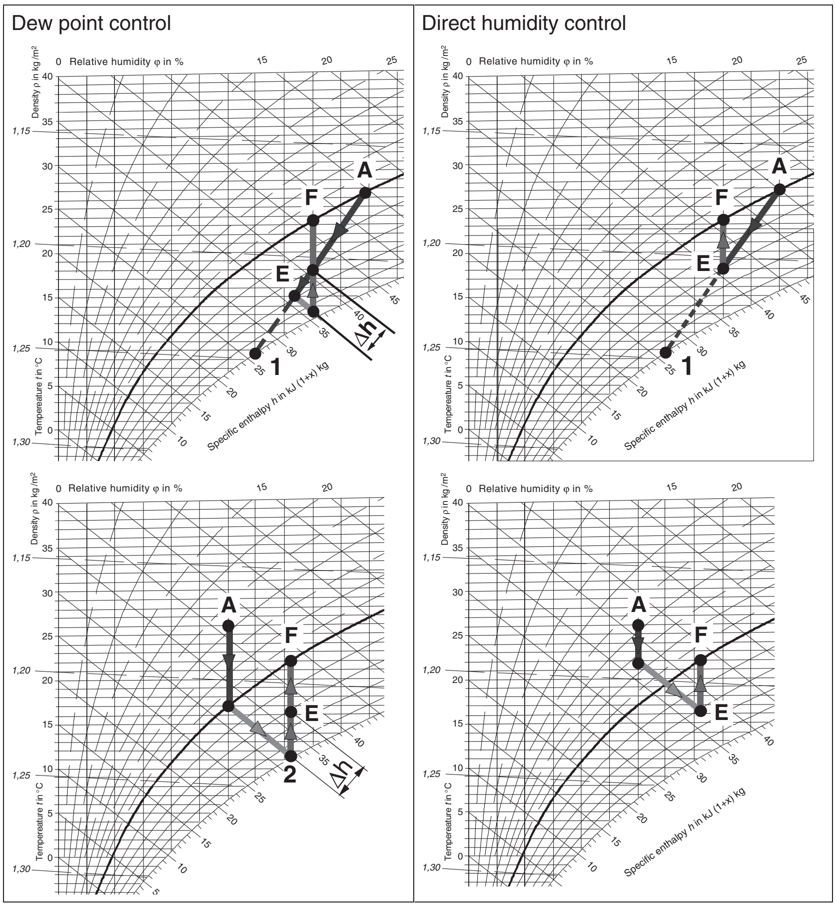


Fig. 2-15 Dew point control a) and direct humidity control b) comparison

- A Outside air state
- F Desired room air state
- E Necessary supply air state
- 1 Mean cooling coil surface temperature
- 2 Room air dew point

2.4 Two-position humidification control

Where controlled humidification is required in simple systems, it can be achieved using a room hygrostat (two-position controller). If the room humidity falls below the setpoint (- switching differential) the hygrostat activates the humidifier. When the upper switching point of the hygrostat is reached, the humidifier is deactivated again (Fig. 2-16). This type of humidification control can, however, only satisfy relatively modest requirements, because fluctuations of the room humidity are unavoidable with the on/off type of control.

2.4.1 Humidification with an air washer

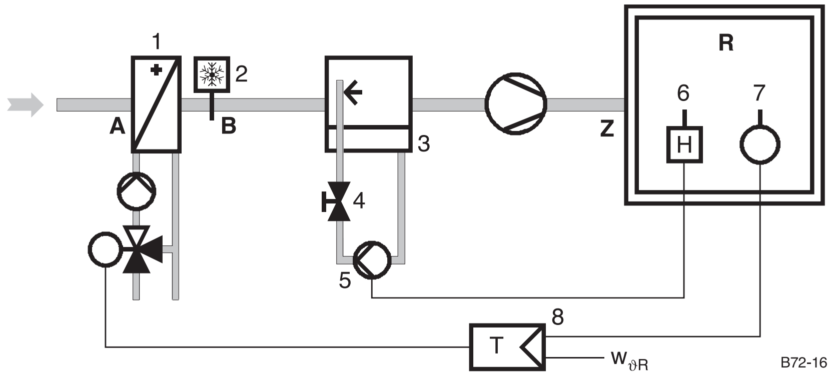


Fig. 2-16 Ventilation system with two-position control of adiabatic humidification
1 Air heating coil
2 Frost protection thermostat
3 Humidifier (e.g. air washer)
4 Adjusting valve (humidification efficiency)
5 Humidification pump
6 Humidity controller (hygrostat)
7 Room temperature sensor
8 Room temperature controller
A Outside air state
B Air state downstream from the air heating coil
Z Necessary supply air state
R Desired room air state

Possible humidifier operating states (Fig. 2-17):

- If the air inlet state A is on or below the adiabatic curve of the minimal necessary supply air state Z, i.e. in the horizontally hatched area, the humidifier pump is permanently commanded on by the hygrostat. The humidifier efficiency η_B is adjusted for full load (i.e. to the maximum quantity of water to be absorbed by the air (Δx_{MAX}) by one time throttling of the water circulation volume. The system functions perfectly in this operating state
- However, if the water content x of the inlet air is greater than needed (in the vertically hatched area), the pump remains permanently off
- In between these two boundary areas (dotted area), the hygrostat acts as a controller and switches the humidifier pump on and off intermittently. The control factor of the pump depends on the water vapor content to be made up Δx and on the humidification efficiency set

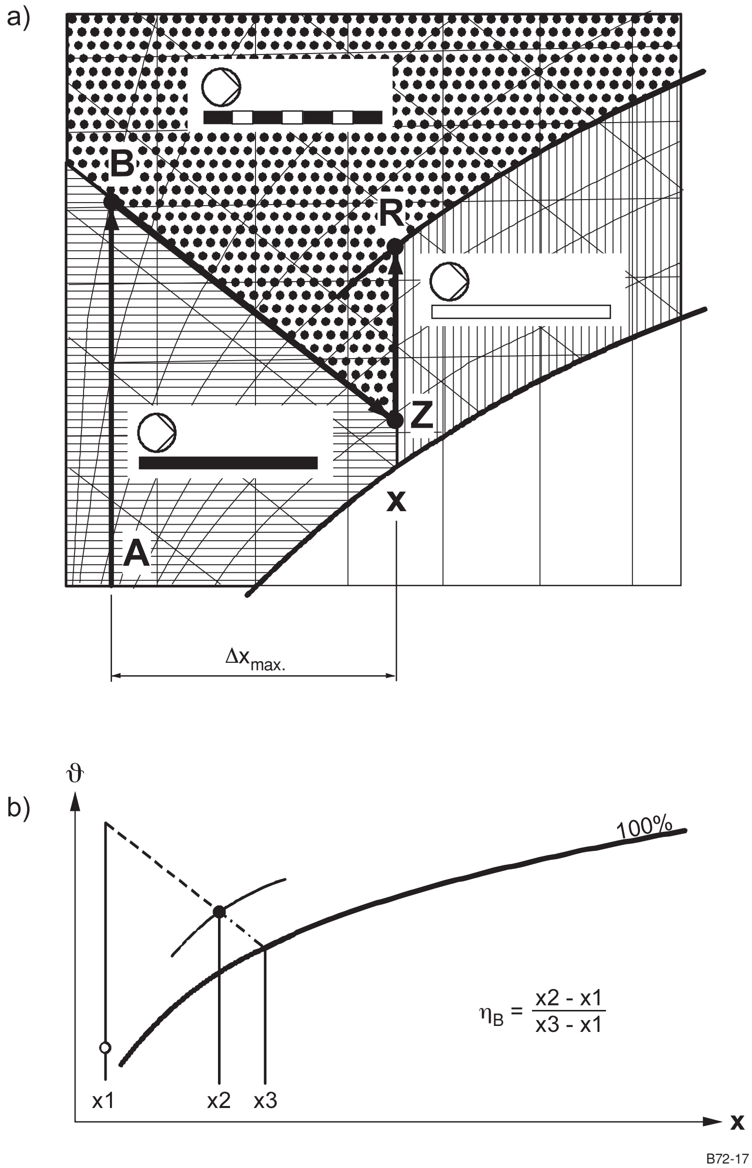


Fig. 2-17 Two-position humidification control with an adiabatic humidifier
a) Operating states of the humidifier pump
b) Humidification efficiency (see also Fig. 2-10b)
Dotted area: intermittent operation
Horizontally hatched: pump continually on
Vertically hatched: pump continually off
A,B,Z,R = air states
Δx_{MAX} = maximum required increase in water vapor content x

The problem with intermittent operation of an adiabatic humidifier is that the air is immediately cooled when the humidifier is switched on. The resultant room temperature drop causes the temperature controller to open the heating valve until the room temperature reaches the set-point again. When the hygrostat switches the humidifier off again, the adiabatic cooling of the supply air ceases to occur. As a consequence, the supply air then enters the conditioned space with too high a temperature until the temperature controller closes the heating valve accordingly. In the case of intermittent humidifier operation, therefore, the two-position action of the hygrostat influences the room temperature control, which also fluctuates.

2.4.2 Two-position control with steam humidification

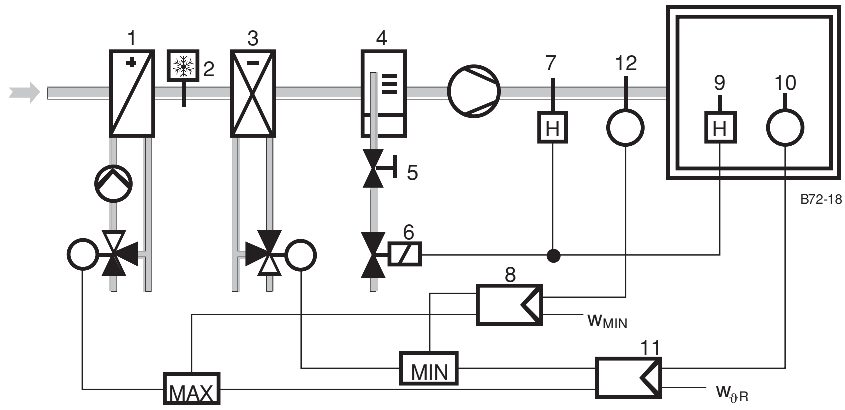


Fig. 2-18 Ventilation system with two-position control of steam humidification

1 Air heating coil
2 Frost protection thermostat
3 Air cooling coil
4 Steam humidifier
5 Throttling valve
6 Humidifier valve (on/off)
7 High limit control hygrostat
8 Supply air temperature low limit controller
9 Room hygrostat
10 Room temperature sensor
11 Room temperature controller
12 Supply air temperature sensor/supply air limit sensor

Fig. 2-18 shows a ventilation system with two-position control of steam humidification. With steam at a temperature of 100 °C, the state change on the psychrometric chart takes place roughly parallel to the isotherm. Therefore, only minor heating but very effective humidification of the air occurs. The hygrostat's control action has (in contrast to adiabatic humidification) practically no effect on the temperature control. In this case however, there is the risk of condensation in the supply air duct due to oversaturation of the air. This risk can be reduced by permanent throttling to limit the maximum permissible quantity of steam to the amount required to humidify to the desired room air state at full load. The risk is even smaller if the permissible quantity of steam is limited to a level which will not give rise to oversaturation even at the lowest permissible supply air temperature (supply air temperature low limit control). Additionally, a high limit hygrostat should be installed in the supply air duct in order to close the humidifier valve when the setpoint is exceeded. Permanent throttling of the steam quantity also provides for an optimal control factor of the humidifier, giving rise to the smallest possible humidity fluctuations in the conditioned space.

3. Recirculated air mixing

3.1 General

In order to improve the economy of ventilation and air conditioning systems, a portion of the exhaust air (its hygienic quality permitting) can be mixed with the fresh outside air. The mixed air is then returned to the conditioned space after further treatment (filtration, heating or cooling etc.).

Mixing of warm exhaust air from the conditioned space can considerably reduce heat energy consumption in winter in comparison to operation with outside air only. In summer, if cooling is present, cooler extract air from the conditioned space is mixed with warm outside air, which considerably reduces the amount of cooling energy required.

Recirculated air mixing is achieved using continuously variable outside air, exhaust air and recirculated air dampers. During cooling operation of the air-conditioning system, the outside air and exhaust air dampers remain fully open and the recirculated air damper closed until the outside air temperature exceeds the extract air temperature. When this occurs, the quantity of outside air is immediately reduced to the required minimum (e.g. as per VDI directives for ventilation, DIN 1946 part 1). If the supply air temperature falls below the corresponding setpoint, the outside air and exhaust air dampers are continuously closed and the recirculated air damper opened until the minimum outside air rate is reached. Since the flow volume through the dampers does not vary in a linear relationship to the closing angle, the minimum outside air rate should be set according to measured values. The minimum permissible outside air rate can normally be set at the temperature controller in the form of a low limit of the respective positioning signal.

Recirculated air mixing can be controlled according to various criteria, such as:

- constant recirculated air mixing
- manual control
- outside temperature-dependent open loop control*
- based on the actuating signal of a mixed air temperature or mixed air enthalpy controller
- based on the actuating signal of a supply air temperature sequence controller with the actuating sequences: heating valve – cooling valve – dampers*

* (possibly in combination with a two-position switchover to the maximum recirculated air quantity during cooling operation, as soon as the outside air temperature or enthalpy rises above that of the extract air.)

3.2 Constant recirculated air mixing

Constant recirculated air mixing is the simplest mode of operation of mixing dampers. The dampers are set to a predefined position so that a certain proportion of recirculated air is mixed with the outside air (the recirculated air quantity is normally selected in order to comply with minimum outside air requirements). Naturally, this proportion affects the design of the ventilation or air conditioning system. The greater the quantity of recirculated air the less power is required for air, heating in winter and for cooling in summer. Constant recirculated air mixing is used in systems with practically constant internal heat gains.

3.3 Manual control of recirculated air mixing

In this case, the desired recirculated air quantity is increased or reduced continuously using an adjusting potentiometer or in steps using a selector switch depending on the outside temperature or the occupancy level of the conditioned space (e.g. conference rooms, dining halls, sports halls). In terms of energy consumption, however, this type of control is not recommended, because the reduction of the outside air quantity is usually forgotten when the occupancy level decreases again.

3.4 Open loop damper control during heating operation

With this type of control (Fig. 3-1), the position of the outside air, exhaust air and recirculated air dampers is varied in proportion to the outside air temperature during winter operation, i.e. in order to reduce heat energy consumption, the recirculated air quantity is continuously increased as the outside temperature falls until the minimum outside air quantity is reached at a certain outside temperature.

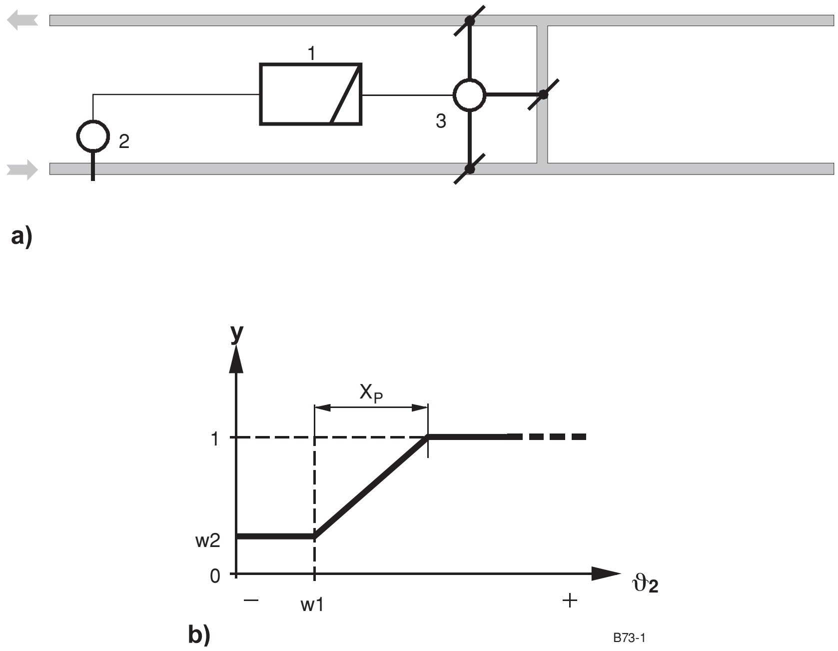


Fig. 3-1 Outside temperature-dependent recirculated air mixing control
a) Schematic
b) Function diagram
1 Damper controller
2 Outside air temperature sensor
3 Damper actuator
ϑ_2 Outside air temperature
w_1 Base value (heating)
w_2 Damper limit value (minimum outside air quantity)
X_P Controller proportional band
y Manipulated variable (outside air damper)

The controller (1) acquires the outside temperature via the sensor (2), Fig. 3-1a. If the outside temperature falls into the proportional band X_P (Fig. 3-1b), the damper actuator (3) begins to close the outside air and exhaust air dampers and open the recirculated air damper in proportion to the outside temperature. The outside air damper is only closed to the selected minimal position w_2 in order to comply with the required minimum outside air quantity.

In terms of energy consumption, this type of control is not optimal because the non-linearity of the damper characteristics and changes in the recirculated air temperature with varying room temperature are not accounted for. Additionally, controlled free cooling in case of internal heat gains (injection of an appropriate volume of cooler outside air) is not possible. Therefore, this type of control should only be used if no internal heat sources are present and if there is no mixing section downstream from the mixing dampers to ensure full mixing of the cold and warm air.

3.5 Open loop damper control during cooling operation

In order to reduce cooling energy consumption in systems with air cooling, the outside air is reduced to the minimum required amount when a definable temperature limit w_3 is exceeded during cooling operation (schematic Fig. 3-1a, function diagram Fig. 3-3). The position change of the exhaust air damper is synchronized with the outside air damper, and the recirculated air damper is opened accordingly.

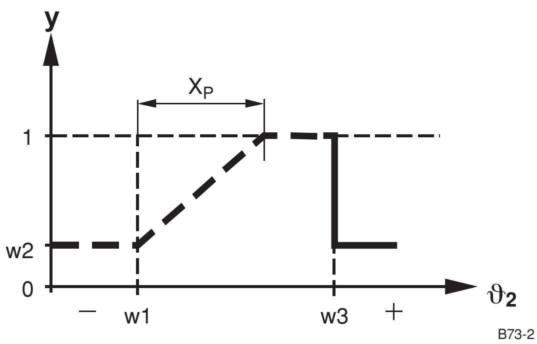


Fig. 3-2 Outside temperature-dependent damper closing function in summer operation

- ϑ_2 Outside air temperature
- w_1 Base value (heating)
- w_2 Damper limit value (minimum outside air quantity)
- w_3 Temperature limit for changeover to minimum outside air quantity in cooling mode
- X_P Controller proportional band
- y Manipulated variable (outside air damper)

The disadvantage of outside temperature dependent on/off operation during cooling is that changes of the recirculated air temperature (e.g. in case of room temperature control with outside temperature compensation) are not accounted for.

3.6 Energy-optimized damper control based on the temperature or enthalpy difference between extract air and outside air

The disadvantage referred to in 3.5 can be avoided if a temperature or enthalpy difference switchover is added to the system (Fig. 3-3a).

3.6.1 Heating operation

The controller (1) acquires the outside temperature via the sensor (2). If the outside temperature falls, it closes the outside air and extract air dampers within the defined proportional band X_P and opens the recirculated air damper (function as described in 3.4).

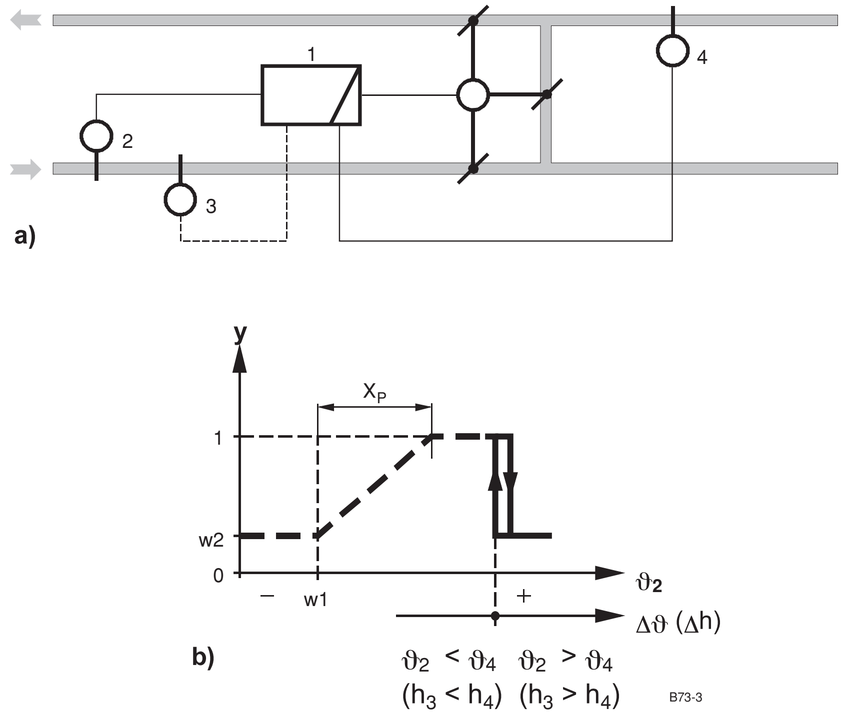


Fig. 3-3 Energy-optimized damper control

a)	Schematic
b)	Function diagram
1	damper controller
2	Outside air temperature sensor
3	Outside air enthalpy sensor (in case of (h switchover only)
4	Extract air temperature sensor or enthalpy sensor
ϑ_2	Outside air temperature
ϑ_4	Extract air temperature
$\Delta\vartheta$	Temperature difference
h_3	Outside air enthalpy
h_4	Extract air enthalpy
Δh	Enthalpy difference

3.6.2 Cooling operation

During cooling operation, the cooling of a large volume of outside air that is warmer than the extract air (or has a higher enthalpy), should be avoided for economical reasons. Therefore, the outside air and extract air dampers are closed to the minimum position by an on/off control action if the outside air temperature acquired by sensor (2, or the outside air enthalpy acquired by sensor 3) is higher than the extract air temperature (or extract air enthalpy) acquired by sensor (4). The corresponding function diagram is shown in Fig. 3-3b.

The advantage of this type of control compared to on/off control based on the outside temperature alone is that the upper closing point of the outside air damper is varied according to the extract air temperature (influence of the outside temperature shift controller on the room temperature, room temperature setpoint adjustment). This gives rise to even more economical system operation. Therefore, this control type is recommended for systems with cooling and shifting room temperature control.

3.7 Energy optimized recirculated air mixing based on mixed air temperature or mixed air enthalpy

In the case of mixed air control (Fig. 3-4), the controller (1) compares the mixed air temperature or enthalpy acquired by the sensor (2) against the selected setpoint (w). The controller adjusts the dampers via the actuator (3) in proportion to the control deviation, causing the mixing ratio of recirculated air and outside air to change. If the mixed air state is below the setpoint selected at the controller, for example, the outside air damper (4) is closed (less cold air) and the recirculation damper (5) is opened (more warm air) until the setpoint is achieved. The dampers are closed only to the selected minimum position in order to comply with the required minimum outside air quantity.

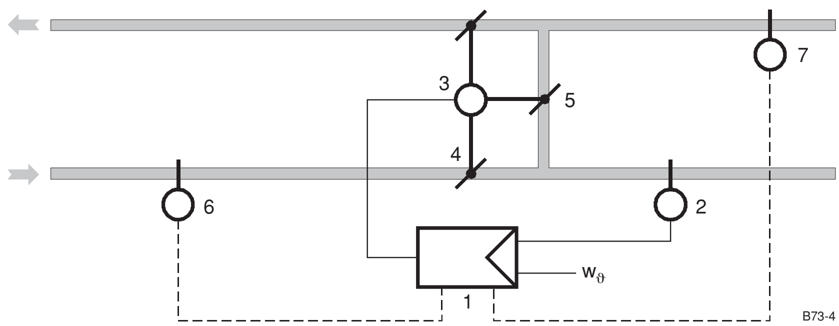


Fig. 3-4 Mixed air control
1 Temperature or enthalpy controller
2 Mixed air temperature or enthalpy sensor
3 damper actuator
4 Outside air damper
5 Recirculation damper
6 Outside air temperature or enthalpy sensor
7 Extract air temperature or enthalpy sensor

Mixed air control reduces the disadvantages of open loop outside temperature-dependent damper control for the following reasons:

- Because the mixed air sensor acquires the temperature or enthalpy of both the outside air and the recirculated air via the mixed air state, any change of either of these variables will be corrected by a corresponding adjustment of the dampers
- The damper characteristics are linearized to a certain extent in the positioning range, because the controller always adjusts the dampers until the required mixed air state is achieved

An energy-optimized damper switchover can be added to the mixing control by connecting 2 additional sensors to the controller (Fig. 3-4). Independent of the control function described above, this will close the outside air and exhaust air dampers to the minimum position (Fig. 3-5) if the outside temperature (or enthalpy) acquired by sensor (6) is higher than the extract air temperature (or enthalpy) acquired by sensor (7). This switchover to the minimum outside air quantity gives rise to a reduction in cooling energy consumption.

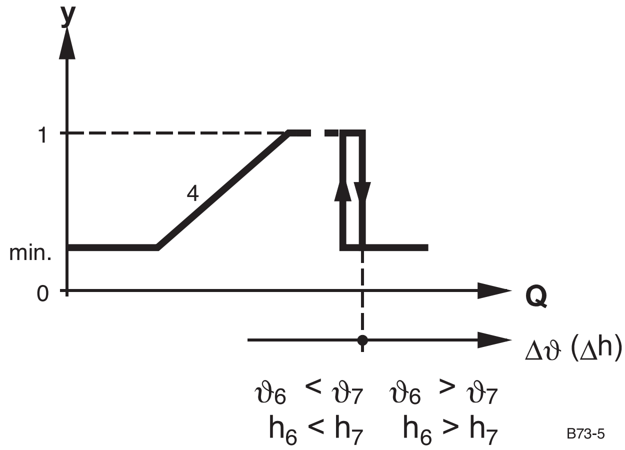


Fig. 3-5 Function diagram, mixing control with maximum economy changeover

Q Load

$\Delta\vartheta$ Temperature differential, outside air (ϑ_6) – extract air (ϑ_7)

Δh Enthalpy difference, outside air (h_6) – extract air (h_7)

4 Outside air damper positioning signal

3.8 Sequence control

In ventilation systems in which the required supply air temperature is to be achieved partly by mixing recirculated air and partly by heating in an air heating coil, the heating valve can be controlled sequentially with the air dampers (Fig. 3-6). A distinction is made between comfort control and economy control depending on whether the outside air damper is closed first or the heating valve is opened first.

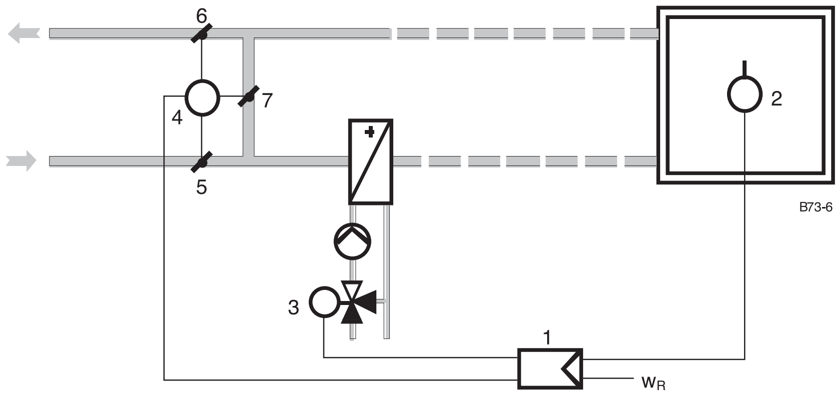


Fig. 3-6 Sequential control of air dampers and heating valve
1 Room temperature controller
2 Room temperature sensor
3 Motorized heating valve
4 Air damper actuator
5 Outside air damper
6 Exhaust air damper
7 Recirculated air damper

3.8.1 Comfort control

The modulating temperature controller (1) compares the temperature acquired by the room temperature sensor (2) against the setpoint. In case of a deviation, the controller first opens the heating valve (3, control function diagram Fig. 3-7). If the air heating coil output is not sufficient although the heating valve is fully opened, the controller closes the outside air and exhaust air dampers (5 and 6) and opens the recirculated air damper (7) via the damper actuator (4). The amount of warm exhaust air necessary to maintain the required room temperature (or supply air temperature) is now automatically mixed with the outside air. The minimum air damper position to ensure a minimum outside air rate can be set on the controller.

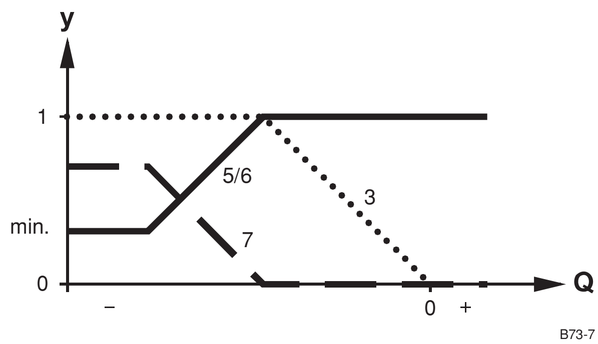


Fig. 3-7 Comfort control function diagram
3 Heating valve
5/6 Outside/exhaust air damper
7 Recirculated air damper

In the case of comfort control, therefore, the heating valve is opened first before the outside air damper begins to close. This keeps the outside air quantity at 100 % for as long as possible, which certainly provides comfort from the hygienic point of view. Since, however, the air heating coil will be sized such that its output will only be insufficient without recirculated air mixing on very few days during the heating season, the system will be operated with 100 % outside air for the major part of its operating time. This gives rise to a correspondingly high heating energy consumption.

3.8.2 Economy control

If the controlled variable falls below the setpoint of the controller (1), first the outside air and exhaust air dampers (5 and 6) are closed and the recirculated air damper (7) is opened simultaneously (Fig. 3-8). The controller does not begin to open the heating valve (3) until the outside air and exhaust air dampers have reached the minimum position. Apart from the actual heating load of the room, the air heating coil only has to heat up the minimum outside air quantity. Compared to comfort control, this requires a considerably smaller air heating coil output and, depending on the climatic zone, the heating valve must only be fully opened on very few days during the heating season. Therefore, this type of control provides a lower rate of room air exchange than comfort control, but is considerably more economical.

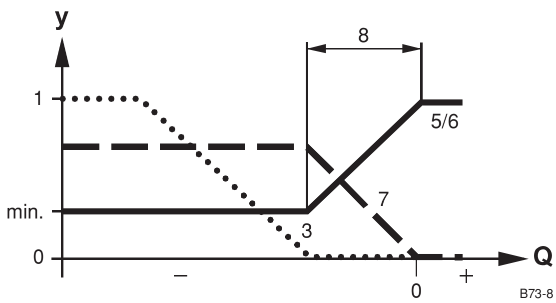


Fig. 3-8 Economy control function diagram

3 Heating valve

5/6 Outside/extract air damper

7 Recirculated air damper

8 Controlled free cooling

In the case of room temperature control, it is also possible to reduce the cooling energy consumption in systems with internal heat sources. In the outside temperature range from approximately +10...20 °C, the room temperature controller can position the outside/recirculated air dampers such that the internal heat can be dissipated by an appropriately increased outside air quantity (controlled free cooling, 8). If the heating load in the conditioned space is zero because of internal heat sources or a rise in the outside temperature, the outside air can be injected into the conditioned space without heating and, provided it is cooler than the room temperature setpoint, also without cooling.

3.8.3 Sequential control, air dampers – heating/cooling valve

The ventilation system shown in Fig. 3-9 is additionally equipped with an air cooling coil. For the purpose of energy optimization, only the air with the lower temperature (outside air or recirculated air) should be cooled during cooling operation. The switchover of the air dampers between 100 % and the minimum quantity of outside air is achieved via a temperature difference switchover (10) which has a temperature sensor in the outside air and one in the extract air and which acts on the air damper positioning signal of the temperature controller. In more complex systems, an enthalpy difference switchover can also be achieved using appropriate sensors (Fig. 3-10).

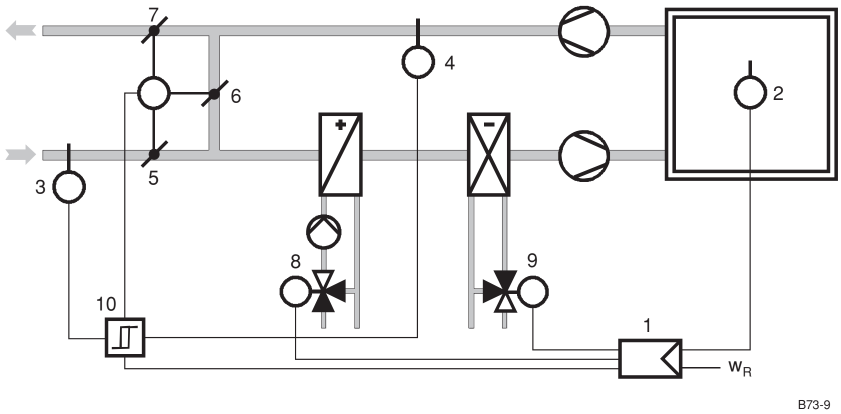


Fig. 3-9 Sequential control, air dampers – heating valve – cooling valve

1 Temperature controller
2 Room temperature sensor
3 Outside air temperature or enthalpy sensor
4 Extract air temperature or enthalpy sensor
5 Outside air damper
6 Recirculated air damper
7 Exhaust air damper
8 Motorized heating valve
9 Motorized cooling valve
10 MAX/MIN changeover of the air dampers according to the temperature differential

The modulating temperature controller (1) acquires the room temperature via the sensor (2). If it falls below the setpoint (Fig. 3-10), the controller closes the outside air and exhaust air dampers (5 and 7) and subsequently opens the heating valve (8). The air dampers can only close to a minimum position in order to ensure that a minimum outside air rate remains.

If the room temperature exceeds the setpoint, the controller first opens the outside air and exhaust air dampers and subsequently the cooling valve.

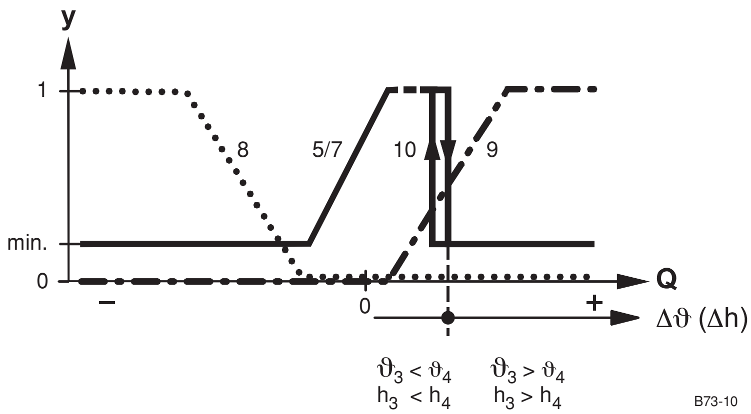


Fig. 3-10 Control sequence diagram for: air dampers – heating valve – cooling valve with temperature or enthalpy difference changeover

Q Load (– = heating load, + = cooling load)

$\Delta\vartheta$ Temperature difference, outside air (ϑ_3) – extract air (ϑ_4)

(Δh) Enthalpy difference, outside air (h_3) – extract air (h_4)

5/7 Outside/exhaust air dampers

8 Heating valve

9 Cooling valve

10 Temperature or enthalpy difference changeover

Independent of the control function described above, the outside air and exhaust air dampers are closed to the minimum position if the outside temperature or enthalpy acquired by sensor (3) is higher than the extract air temperature or enthalpy acquired by sensor (4). In this way, the ventilation system operates with a minimum outside air rate and the cooling energy consumption is reduced.

If the outside air temperature or enthalpy falls below the extract air temperature or enthalpy again, the temperature or enthalpy difference changeover switches back to the air damper actuation signal of the controller, i.e. the air dampers are again connected in sequence with the heating valve to the room temperature control.

3.8.4 Indoor air quality control

Objective
Improved comfort at low operating costs

With this control strategy, the ventilation plant is controlled on the basis of the indoor air quality, with the aim of improving comfort while keeping the cost of operating the plant as low as possible.

Indoor Air Quality (IAQ)

The abbreviation IAQ (Indoor Air Quality) is an internationally accepted abbreviation. It was used primarily by the World Health Organization (WHO) and the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) and therefore appears in the relevant standards and directives.
The use of a demand-based air quality control system ensures indoor comfort by taking account of the indoor air quality as a control variable in the control strategy.

Benefits

The benefits of demand-based air quality control are:

- Good air quality reduces SBS (Sick Building Syndrome)
  ⇨ Increases staff productivity, reduces absences due to ill-health
  An increase of as little as 1 % in productivity can justify a 50 % increase in the capital investment for heating and ventilation systems. Studies have shown that office employees affected by SBS work up to 10 % more slowly and make up to 30 % more errors.
- Concentration improves when the CO_2 content in the air is sufficiently low ⇨ Important in classrooms and office buildings
- People spend more time in restaurants and shopping centers with good air quality ⇨ Increased turnover
- The plant operates automatically on the basis of the indoor air quality ⇨ No need for staff to think about the system, no problems with ventilation switched on too late or left on for too long (e.g. in restaurants)

Energy cost savings

Significant energy cost savings can often be made (30 % to 50 % in some cases, based on evidence from case studies). This is primarily explained by two factors:

- The plant only operates when necessary
  ⇨ Savings made in relation to the conditioning of the outside air (e.g. heating or cooling coils)
- The power consumption of the fans is reduced substantially (e.g. by speed control or modulating control via a variable speed drive) since power consumption varies as the cube of air volume (cf. Fan laws).

Measured variables
CO_2, VOC

Air quality is measured by two different variables:

- **CO_2**
  (carbon dioxide)

- **Mixed gases / VOC**
  (Volatile Organic Compounds)

CO_2 (carbon dioxide)

CO_2 is an odorless gas which is the product of combustion (e.g. heating oil, petrol) and also of respiration. CO_2 is a significant variable in relation to indoor air quality, because human beings produce CO_2 when they exhale, thereby increasing the concentration of CO_2 in occupied spaces ⇨ CO_2 concentration in a space is an indicator of the number of people in the space

CO_2 also has health implications (our breathing is controlled by CO_2 content) and hence our sense of well-being and ability to concentrate are affected when certain ppm levels are exceeded:
CO_2 is measured in volume % or ppm (parts per million)
The CO_2 content in normal outdoor air is approximately 0.033 volume % or 330 ppm.

Concentration [ppm]	
330	Concentration in normal outdoor air
1 000	Upon entering a room, 20 % of people will be dissatisfied with the prevailing air quality
1 500	Limit value specified in DIN 1946
2 000	Sensitive persons complain of headaches
4 000	Maximum value in a classroom at the end of lessons
5 000	MAC value (Maximum Acceptable Concentration in the workplace)
100 000	Sufficient to extinguish a candle in the room and cause loss of consciousness

Fig. 3-11 CO_2 concentration levels and limit values

Measuring principle

The CO_2 sensor is based on photo-acoustic measurements. The principle was discovered as early as 1880 by A.G. Bell and J. Tyndal.
Only three separate components are required for this measuring principle:

- A source of infrared radiation (IR)
- A pressure-insulated measuring cell
- A microphone with signal conditioning

A miniature bulb shines through an optical filter into the measuring cell. The filter only allows light in the infrared range (with a wavelength of around 4.26 µm) to pass through. The CO_2 molecules are stimulated by light at this wavelength, absorb the radiated energy and release it again as they collide with other molecules. This causes a slight increase in pressure inside the measuring cell. If the light-source is periodically switched off, the pressure differences can be detected by a sensitive microphone and evaluated electronically, i.e. converted into the measured signal. This measured signal is proportional to the CO_2 content. An advantage of this method of measurement is that there is no need for regular recalibration of the zero point.

The advantages of the photo-acoustic measurement principle over other methods are as follows:

- Measured signal and the CO_2 concentration are directly proportional
- Highly sensitive method
- Zero point very stable (relative measurement, no calibration necessary)

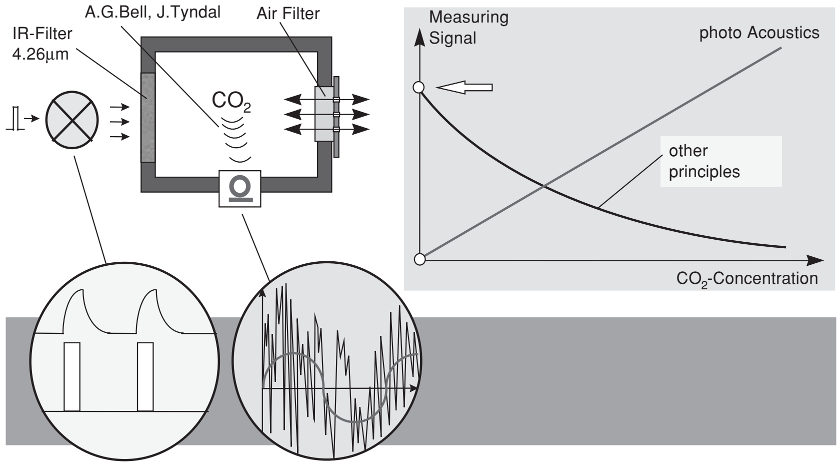


Fig. 3-12 CO_2 measurement based on the principle of photo-acoustic measurement

CO_2 or CO?

CO_2 *must not be confused with CO* (carbon *monoxide*), which occurs as the result of incomplete combustion (e.g. in car exhaust gases) and which, unlike CO_2, is toxic even in very low concentrations. CO must be measured in all areas where it could occur at dangerous concentration levels (e.g. indoor car parks). These measurements, then, are part of a warning system rather than a control system.

Mixed gases / VOC

Sources of mixed gases / VOC are:
- People, especially smokers
- Emissions from furnishings and fabrics etc.
- Ventilation systems, especially if these are poorly maintained

Mixed gases (odors) are much harder to measure than CO_2. There is no sensor in the world which can fully replace the human nose. Volatile organic compounds (VOC) are primarily responsible for poor air quality. Difficulties that arise with the measurement of VOC are:
- Different weightings are applied to different odors (such as tobacco smoke, or emissions from furnishings and fabrics), and these do not necessarily correspond to the experience of the human nose.
- The measurement depends on a comparison between the indoor and outdoor air ⇨ This is a problem if the outdoor air quality is poor.

Without further signal processing, however, the commercially available VOC sensors are not sufficiently accurate or reliable for use in air conditioning applications, as we need to be able to measure even the smallest concentrations of VOC. However, with an intelligent algorithm, the VOC concentration can be adapted to the current quality of the supply air, and can then measure even very low concentrations of VOC reliably and stably.

Measuring principle

VOC levels are detected using a gas sensor based on the Taguchi principle (Fig. 3-13). Essentially, this consists of a sintered ceramic tube containing a filament. Flat-panel thick-film technology sensors have been developed in recent years. The sensor tube is highly porous and thus has a very large surface area. It consists of doped tin dioxide (SnO_2). This sensitive material works on the redox principle.

When the sensor tube is heated to 400 °C, free electrons form on the sensor surface. These attract oxygen atoms, which settle on the surface. Ideally, any gases or vapors coming into contact with the sensor surface will be oxidized, giving CO_2 and water vapor. In this process, the oxygen required for the oxidation process is removed from the sensor surface. This frees electrons, thereby changing the resistance of the semiconductor. This change in resistance can be measured. The oxygen used in combustion is replaced by oxygen from the air. The mixed gas sensor can thus detect all gases capable of oxidation in proportion to their concentration and redox potential.

The Taguchi principle of measurement has long been used with success for the detection of gas leaks. It has rarely been used for the measurement of air quality in HVAC systems. One reason, certainly, is that the required measuring range is at the lowest sensitivity zone of the Taguchi sensor, where the base value of the sensor signal is relatively widely scattered and subject to drift. Another reason is that, except in relation to CO_2, there is no technical literature with quantifiable and binding statements in relation to odor and air quality.

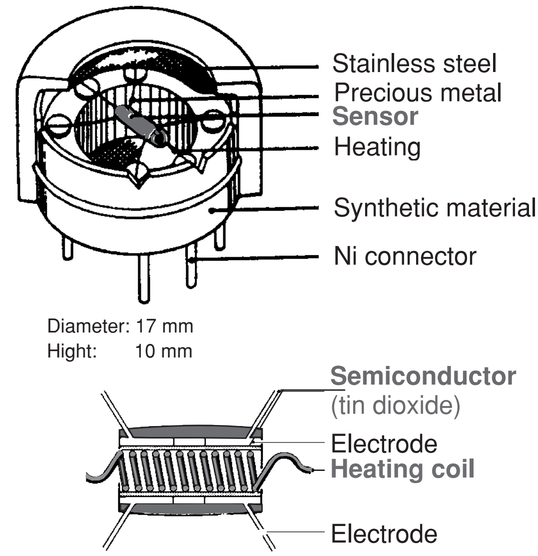


Fig. 3-13 VOC measurement with gas sensor based on the Taguchi principle

Increased use of combined CO_2/VOC measurement

Today, combined CO_2/VOC measurements are seen as the optimum solution for air quality control.

Other indicators

In relation to odor, the terms *olf, decipol and percentage dissatisfied* (PD) are used, derived from the Fanger's theories). These variables cannot be measured like CO_2, but they do indicate the relationship between odor, the number of air changes and the degree to which occupants of a space find the air quality satisfactory. They cannot be used for control purposes, but are useful e.g. when defining setpoints (cf. Fig. 3-11, value for 1000 ppm CO_2).

olf

1 olf is equivalent to the air pollution caused by a "standard person"
A "standard person" is an adult in a sedentary occupation, with a hygiene standard of 0.7 baths per day. The "olf" unit is used to indicate the pollution load caused by both objects and people. It is not a directly measurable variable: it can only be determined on the basis of the perceived air quality.

decipol

1 decipol represents the perceived air quality in a room with a pollution load of 1 olf and a ventilation rate of 10 l/s or 36 m³/h.
The decipol is the unit of measurement of perceived air quality.

PD = Percentage Dissatisfied

The perceived air quality is defined using data from a number of test persons, whereby the "percentage dissatisfied" is established on the basis of the number who find the air quality unsatisfactory. These values are recorded in relation to the outside air rate, which in turn, is related to the number of people occupying the space.

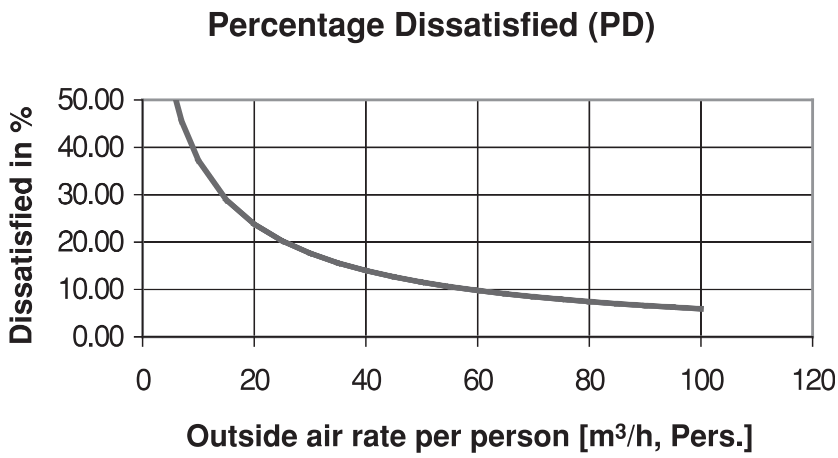


Fig. 3-14 Relationship of Percentage Dissatisfied (PD) to the outside air rate per person

Ventilation demand processor

The ventilation demand processor is a vital element of a demand-based air quality control system. It registers the two sensor signals (CO_2 and VOC signal) and processes them to give a ventilation demand signal. The ventilation demand signal is the result of maximum value selection from the CO_2 signal and, after filtering in the processor, the VOC sensor signal.

The ventilation demand is indicated by a 2...10 V signal, corresponding to a CO_2 range of 400...2,000 ppm. The influence of the VOC ventilation demand on maximum selection in relation to the CO_2-based ventilation demand is a modifiable value equivalent to ± 200 ppm CO_2.

Advantage

Using a separate ventilation demand processor means that its complex algorithm does not need to be repeatedly reprogrammed, and ensures a high standard of air quality control.

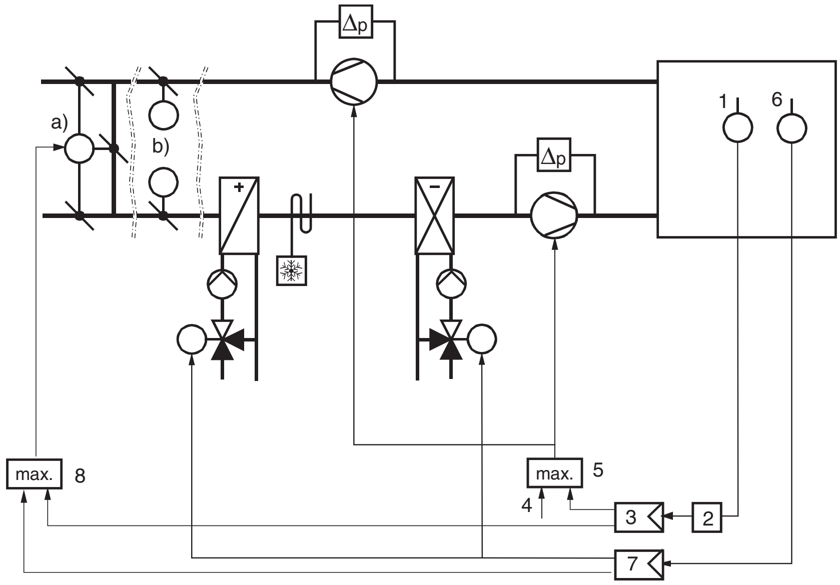


Fig. 3-15 Air quality control combined with temperature control

1 Air quality sensor (CO_2/VOC)
2 Ventilation demand processor
3 Air quality controller
4 Ventilation control demand signal (e.g. from time schedule, occupancy sensor etc.)
5 Max. value selector for fan control demand signals
6 Room temperature sensor
7 Room temperature controller
8 Max. value selector for heating/cooling demand signals
a) Variant with mixing of recirculated air
b) Variant using outside air only

Principle of operation

The air renewal demand is detected with a combined CO_2/mixed gas (VOC) sensor (1). The air renewal demand signal is calculated by a separate ventilation demand processor (2). The ventilation demand signal is transmitted to the air quality controller (3). At the same time, demand signals are received from the fan control system (4), e.g. from time schedules, occupancy sensors etc.). A maximum value selector function (5) always selects the higher of the two signals and transmits this to the fans. The temperature control system uses the room temperature sensor (6) and the temperature controller (7) to determine a demand signal for heating/cooling. This signal is compared with the air quality control demand signal for the recirculated air dampers, and the higher of the two signals is selected (8) and transmitted to the heating or cooling coil.

AQ control strategies

Different air quality control strategies are employed depending on whether the system is fitted with recirculated air dampers (a) or designed for operation with outside air only (b). Three common types of system with the associated control strategies are discussed below. These are:
- System with two-speed fans and direct mixing of the recirculated air
- Outside air system with two-speed fans
- System with speed-controlled fans

These air quality control strategies also need a temperature control loop (e.g. controller 7 in Fig. 3-15, whose function has already been described under 3.8.3).

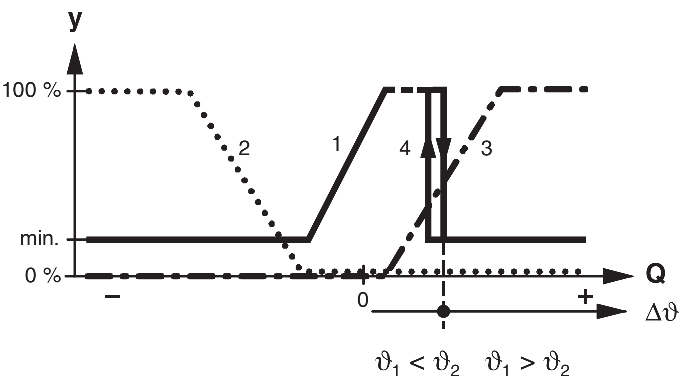


Fig. 3-16 Operating diagram for the circuit incorporating: dampers, heating valve and cooling valve with changeover based on temperature differential

Q Load (– = heating load, + = cooling load)

$\Delta\vartheta$ Temperature differential between outside air (ϑ_1) and extract air (ϑ_2)

1 Outside air/exhaust air damper

2 Heating valve

3 Cooling valve

4 Outside air damper/exhaust air damper changeover based on temperature differential

Plant with two-speed fans and direct mixing of recirculated air
- If the air quality deteriorates, the system initially switches to the first fan stage (I)
- The outside air damper goes to the preset minimum position and is opened proportionally in response to a further deterioration in air quality.
- The second fan speed (II) is not enabled until the damper is fully open
- At low outside temperatures, an upper limit can be applied to the outside air volume (⇨ max.) to ensure that the heating coil can achieve the required supply air temperature

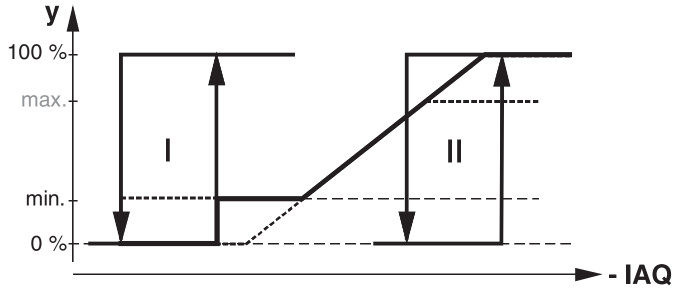


Fig. 3-17 Demand signals for systems with 2-speed fans (I, II) and direct mixing of the recirculated air
y Outside air damper positioning signal
-IAQ Deteriorating indoor air quality

Outside air system with two-speed fans

- If the air quality deteriorates, the system is switched to the first fan speed (I)
- The second fan speed (II) is enabled if the air quality deteriorates further.

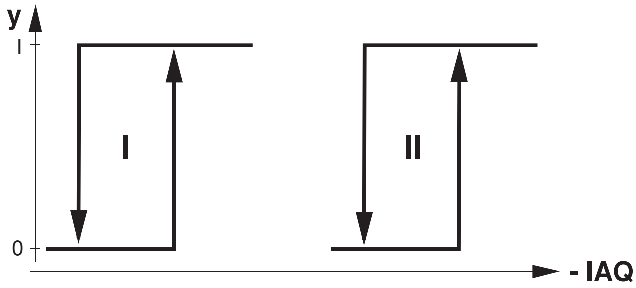


Fig. 3-18 Demand signal for outside air system with two-speed fans (I, II)
y Fan speed control signal
-IAQ Deteriorating indoor air quality

System with speed-controlled fans

- If the air quality deteriorates, the plant is switched on. This enables the fans (I) which run at minimum speed.
- The outside air damper opens 100 %
- If the air quality deteriorates further, the fan speed is increased in proportion (control signal y)

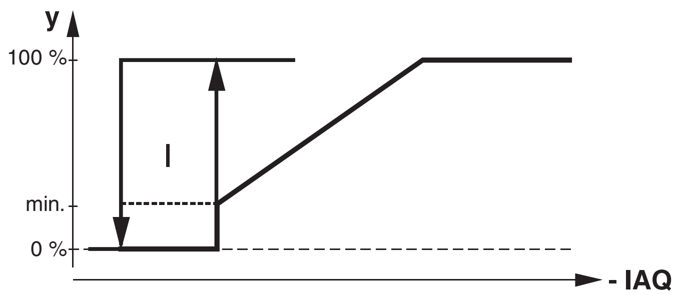


Fig. 3-19 Demand signals for systems with speed-controlled fans
y Fan-speed control signal
-IAQ Deteriorating indoor air quality

Enable strategies

A system which operates using demand-based air quality control can be switched on and off in accordance with various criteria. The diagram below shows a system over 24 hours, with various strategies for enabling/disabling the plant.

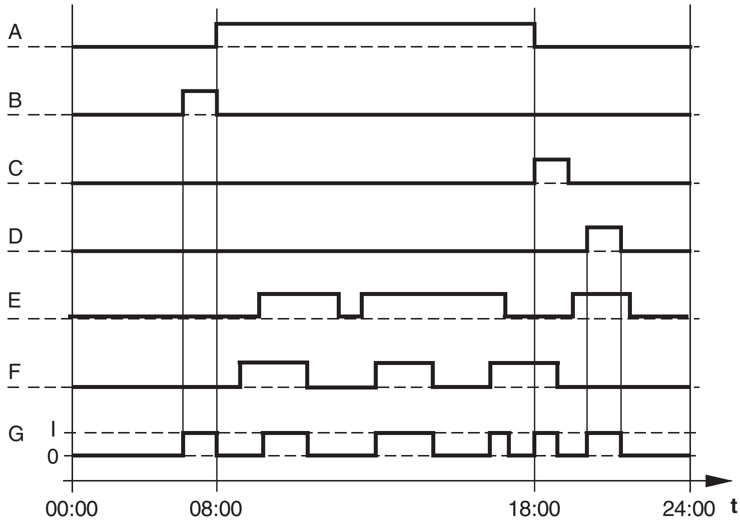


Fig. 3-20 System with time schedule, manual intervention and occupancy sensor

A Overall period of occupancy
B Pre-ventilation
C Post-ventilation
D Manual ON
E Occupancy sensor
F Air renewal demand
G System

Possible reasons for enabling/disabling the plant are as follows:

Overall period of occupancy

Within the overall period of occupancy, there are normally people in the building. Here, the role of demand-based control is to ensure comfort conditions (good air quality).

Pre-ventilation

It is important to ensure good air quality in a space before it is occupied. The duration of the pre-ventilation function (or “purge” prior to occupancy) can be specified (approximate two air-changes), and should end at the start of occupancy. During the pre-ventilation period, the system operates at the highest fan speed with 100% outside air.

Post-ventilation

The post-ventilation function (or “purge” at end of occupancy) can be used to prevent unpleasant odors from accumulating in fabrics and furnishings outside occupancy hours. Post-ventilation starts at the end of occupancy if the air quality sensor indicates an air renewal demand. It ends at the latest after a user-definable period.

Forced ventilation (Manual ON)

Within and outside the period of occupancy, a pushbutton can be used to force the ventilation system on for a limited period of time. During the forced ventilation period, the system runs at the highest fan speed, with 100% outside air.

Occupancy sensor

In rooms which are not in regular use even during the overall occupancy period, occupancy sensors can be used to detect the presence of people. The demand-based ventilation system is only enabled if the occupancy sensor indicates that there are people in the room.

Coordination with the room temperature control system

Operating independently of the demand-based ventilation system, the temperature controller is required to determine whether or not thermal comfort is being achieved, and to enable the system in the event of a heating or cooling demand.

4. Control and anti-icing protection for heat recovery units (HRU)

4.1 General

The treatment of air in ventilation and air conditioning systems consumes energy. The largest portion is used for heating or cooling and, in the case of complete air conditioning, also for humidification and dehumidification of the air. In comfort air conditioning systems, hygienic considerations dictate that, depending on the number of occupants, a certain air change coefficient [h^{-1}] (outside air supplied to the room [m^3/h] / room volume [m^3]) must be achieved in the conditioned space. This means that, in winter, a large volume of heated and possibly humidified air, and in summer, a large volume of cooled and dehumidified air, must be expelled and replaced with outside air. The target in this case is to reclaim as high a proportion of the expended energy as possible.

4.2 Recirculated air mixing

The systems and control possibilities of recirculated air mixing have been covered in detail in chapter 3. Recirculated air mixing alone is not really heat recovery as such (according to VDI 2071) but rather the prevention of unnecessary energy loss via the exhaust air.

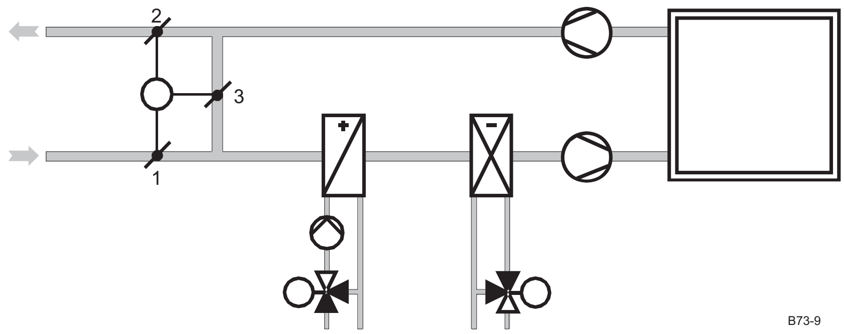


Fig. 4-1 Mixing of the recirculated air
1 Outside air damper
2 Exhaust air damper
3 Recirculated air damper

If the recirculated air quantity must, for hygienic reasons, not exceed about 50 %, it is generally worth combining recirculated air mixing with a heat recovery unit (HRU). Such combinations and the corresponding control functions are covered in 4.8.

4.3 Plate or pipe heat exchangers

The exhaust air and outside air streams are kept apart by fixed separators, normally in the form of cubes divided by plates, but sometimes by pipes. The materials used, e.g. aluminum, chrome steel, glass, plastics, depend on the application and the condition or quality of the air.

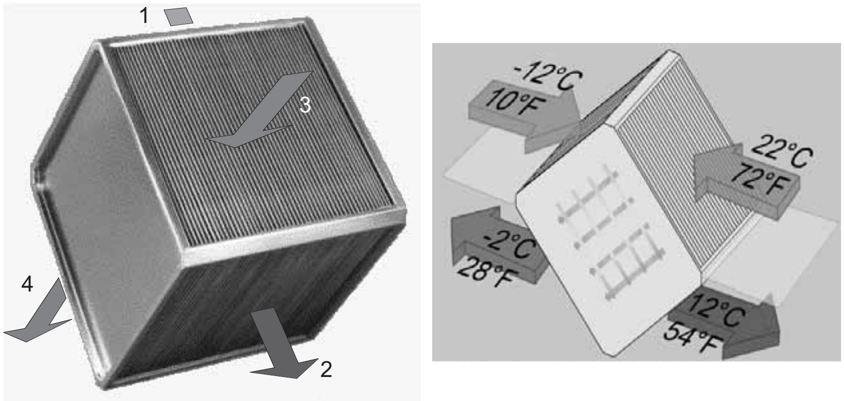


Fig. 4-2 Recuperative heat recovery: Plate heat exchanger (Source: Klingenburg)
1 Outdoor air
2 Supply air
3 Extract air
4 Exhaust air

4.3.1 Control

The heat exchange output of these HRUs can only be regulated via a bypass duct in the extract air or outside air (Fig. 4-3).

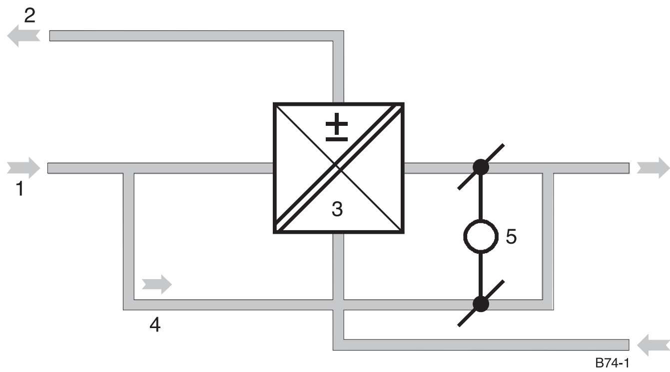


Fig. 4-3 Output control via a bypass duct in the outside air
1 Outside air
2 Exhaust air
3 Heat recovery system
4 Bypass duct
5 Motorized air dampers

The correct sizing of the bypass duct and the installation of opposed-blade dampers are decisive factors for the good controllability of the system. It is recommendable to observe the following rough guidelines:

- The pressure drop across the HRU should not be greater than about 50 % of the pressure loss in the entire ducting
- The pressure drop across the fully open damper should be approximately 10 % of the heat exchanger air resistance
- The pressure drop in the bypass duct should be equal to the pressure loss across the heat exchanger
- Configurations with a single control damper located in the bypass duct and offering too little air resistance give rise to an unacceptable control action

4.3.2 Icing protection

If the temperature of the extract air falls below the dew point at outside temperatures below freezing, ice forms on the extract air side of the heat exchanger, constricting the flow area. This occurrence must be prevented by protective equipment: if extract air mixing is permitted, icing protection can be realized without reducing the degree of change. This circuit is described in 4.8.

If the exhaust air and outside air have to remain completely separated, the anti-icing problem can be solved either with a preheater in the outside air duct or with a bypass duct with control dampers. The first of these solutions is shown in Fig. 4-4 with an electric (or hot water) air heating coil which increases the outside air inlet temperature to an adjustable minimum value.

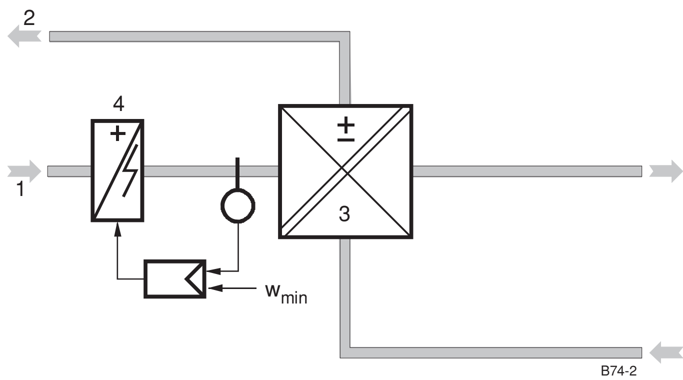


Fig. 4-4 Anti-icing protection control via an electric air heater battery in the outside air duct
1 Outside air
2 Exhaust air
3 Heat recovery system
4 Electric air heater battery with own control loop

The second possibility with the bypass duct is shown in Fig. 4-5. The exhaust air outlet temperature is limited to a minimum value by reducing the outside air flow volume through the heat exchanger accordingly. However, this type of anti-icing protection must be used with caution, because there is no generally applicable value (e.g. 0 °C) for the minimum temperature to be set at the controller. It is specific for each make of heat exchanger and must be determined empirically. It is also conceivable to install the temperature sensor for anti-icing protection at the critical location in the heat exchanger.

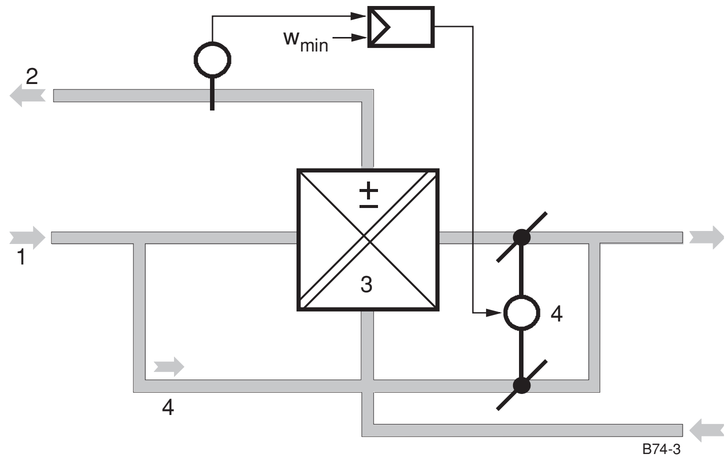


Fig. 4-5 Anti icing protection control via a bypass duct for the outside air
1 Outside air
2 Exhaust air
3 Heat recovery system
4 Air damper control with bypass duct

Instead of a temperature sensor and temperature controller for anti-icing protection, a differential pressure control loop in the extract air duct (with pressure sensors upstream and downstream of the HRU) can be used, since the pressure differential rises with increasing icing of the heat exchanger surfaces. All anti-icing protection controls can be combined with the HRU control by acting on the same control element via a priority control circuit.

Different control variants are shown in Fig. 4-6, with a bypass duct in the extract air and anti-icing protection via a separate control loop with a heat exchanger.

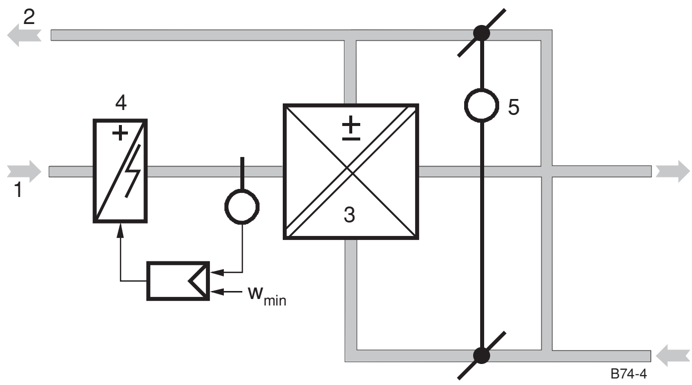


Fig. 4-6 Combination of anti-icing protection and output control via an electric air heater in the outside air duct and a bypass duct for the extract air
1 Outside air
2 Exhaust air
3 Heat recovery system
4 Electric air heater battery with anti-icing protection control loop
5 Dampers with bypass duct for the extract air

Fig. 4-7, with a bypass duct in the outside air and anti-icing protection acting on the damper actuator via a priority control circuit. In the latter case, it is important to ensure thorough mixing of the air flows downstream of the bypass duct. Furthermore, the reduced capacity of the HRU caused by the limit function is often compensated by increasing the output of the downstream heating coil (e.g. Siemens AEROGYR). This method is also used in closed-circuit systems (cf.4.4.2) and rotary heat exchangers (cf. 4.6.2).

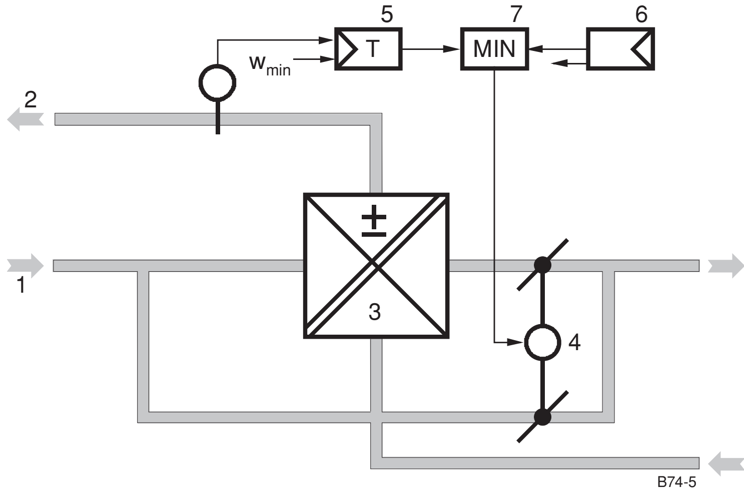


Fig. 4-7 Combination of anti-icing protection and output control via a bypass duct for the outside air and priority control on the damper actuator

1 Outside air
2 Exhaust air
3 Recuperative HRU
4 Air damper control with bypass duct for the outside air
5 Anti-icing protection control
6 System temperature controller
7 Minimum priority selection

The disadvantage of both control types is that the degree of temperature change is reduced by the action of the anti-icing protection control.

4.4 Closed circuit system

4.4.1 Control

Control is best achieved via a three-port valve in the transfer medium (Fig. 4-8). The valve is positioned as close to the outside air heat exchanger as possible and sized according to the usual guidelines. In systems with a wide air flow variation, the maximum transfer medium circulation must be adjusted to the air mass flow rate, e.g. via pump speed control, in order to maintain a water equivalent ratio of 1 so that the system always operates with an optimal overall degree of change.

4.4.2 Anti-icing protection

As soon as the surface temperature of the extract air heat exchanger falls below 0 °C because of the cooled transfer medium, the condensate from the extract air can freeze. This ice formation must be prevented. This is accomplished using the circuit shown in Fig. 4-8, where the cooled transfer medium from the outside air heat exchanger is mixed with an appropriate quantity of warmer transfer medium from the exhaust air heat exchanger (diverting circuit).

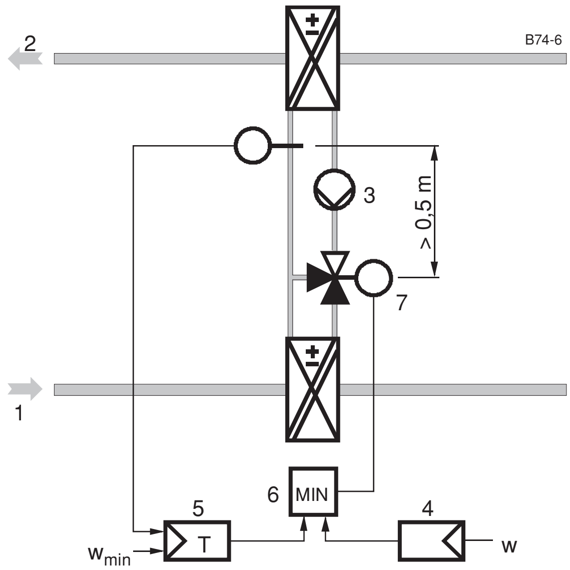


Fig. 4-8 HRU control combined with anti-icing protection control with diverting circuit
1 Outside air
2 Exhaust air
3 Circulating pump
4 HRU output controller
5 Anti-icing protection controller
6 Priority switch
7 Mixing valve

The operating conditions under which there is a genuine icing hazard depend not only on the outside temperature but also on the extract air temperature and on the degree of temperature change of the HRU system.

Fig. 4-9 shows the minimum permissible outside temperature in relation to the degree of temperature change with an extract air temperature of 20 °C. It can be seen from the example plotted that, with a degree of temperature change of 50 %, the minimum permissible outside temperature is approximately –6,5 °C. If anti-icing protection control is required, it can be combined with the output control via a priority circuit (Fig. 4-8).

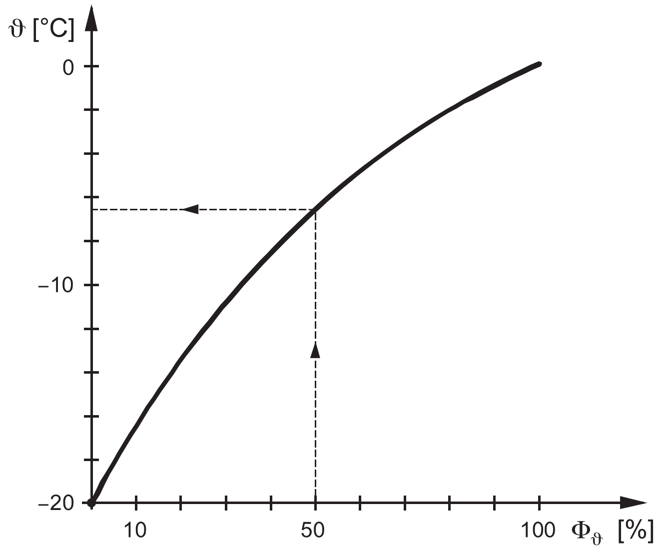


Fig. 4-9 Minimum permissible outside temperature ϑ as a function of the degree of temperature change Φ_ϑ

4.5 Heat pipes

Heat pipes are a refrigerant cycle system without the supply of auxiliary energy. The heat pipe is maintenance-free and also requires no anti-icing protection. Output control is relatively difficult; it can be achieved using a bypass or by varying the angle of inclination via an actuator, for example.

4.6 Thermal wheel type heat exchanger

A rotating cellular thermal storage mass is exposed alternately to an outside air and an exhaust air stream. The amount of heat transferred can be influenced by modifying the rotor speed. The surface of the thermal storage mass can be coated with a hygroscopic substance so that humidity or enthalpy can also be transferred.

In the designs used for ventilation and air conditioning applications, the storage mass is rotated continuously at approximately 1...10 rpm between the outside air and exhaust air. The degree of change depends on the speed of the rotor and the velocity of the inflowing air.

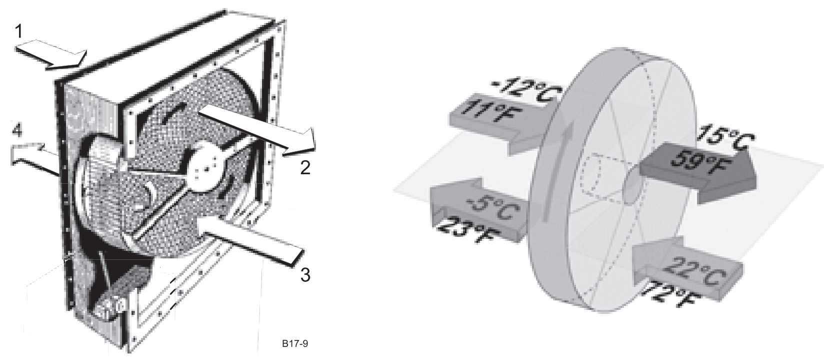


Fig. 4-10 Regenerative heat recovery: Thermal wheel
Cross-section and principle (in summer)
1 Outside air
2 Supply air
3 Extract air
4 Exhaust air

4.6.1 Control

Heat recovery control (Fig. 4-11) is achieved by varying the speed of the rotor. The degree of change rises with increasing speed, until an upper limit speed is reached. The rate of increase depends on the inflowing air velocity; the curve is very steep at low air velocities. Therefore, difficulties (control fluctuations) can be expected in this operating state, because the rotor speed is usually controlled to a low limit in order to keep the rotor clean. This means that the HRU must, on the one hand, be switched off when the low limit is reached and, on the other hand, be periodically started for short times (to clean the rotor). However, this on/off operation can be avoided with the proper design of the HRU.

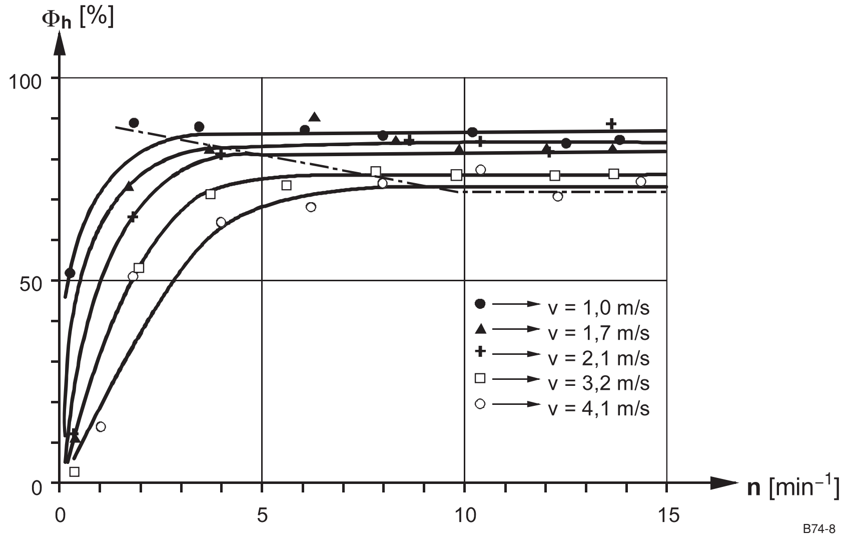


Fig. 4-11 Degree of change of a thermal wheel HRU as a function of rotor speed and inflowing air velocity
- Φ Degree of temperature or enthalpy change
- n Rotor speed (rpm)
- v Inflowing air speed

4.6.2 Anti-icing protection

Ice forms on the rotor below a threshold temperature specified by the manufacturer. The threshold temperature must be maintained either using an air heating coil (Fig. 4-12) or, if permissible, via recirculated air mixing. Anti-icing protection control with recirculated air mixing is covered in 4.8.

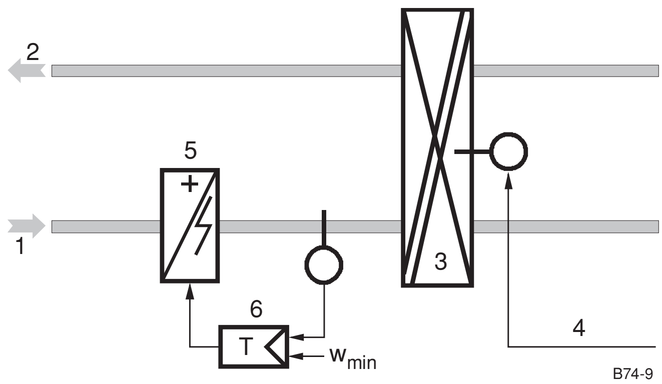


Fig. 4-12 Anti-icing protection control using an electric air heater battery in the outside air duct

1	Outside air	4	HRU control signal
2	Exhaust air	5	Electric air heater battery
3	Thermal wheel type heat exchanger	6	Anti-icing protection control loop

4.7 Heat pumps

Using a heat pump, more heat could be extracted from the extract air of a ventilation or air conditioning system than with the other HRU types. However, the heat pump also has disadvantages in this application:

- Simple heat pumps transport the heat in one direction only. If the direction of heat transport is to be changeable, e.g. for cooling operation of the system, the heat pump must be equipped with an additional reversing circuit
- The system requires relatively high investment and maintenance costs. For these reasons, it is unlikely that the heat pump will become established in the heat recovery systems described here

Therefore, heat pumps are more suitable for heat recovery systems in industrial applications where, for example, relatively constant quantities of waste process heat must be dissipated via a ventilation system.

4.8 Combination of HRU and recirculated air mixing

If an HRU is installed in an air treatment system in which a proportion of recirculated air of approximately 50 % is permissible, it makes sense to increase the amount of heat recovery by combining the HRU with recirculated air mixing. In practice, only the thermal wheel type heat exchanger is appropriate for such a combination, because the closed circuit system and plate heat exchanger are only selected in cases where there must be absolutely no exchange of air between extract air and supply air. This chapter explains the principle system variants, showing the degrees of temperature change that can be achieved with each.

4.8.1 Recirculated air mixing upstream of the HRU

The combination shown in Fig. 4-13 can be used to increase the overall heat recovery on the one hand, and to provide anti-icing protection control for the HRU on the other, provided the necessary minimum recirculated air quantity is permissible in every case. This solution offers the additional advantage that the anti-icing protection can be implemented without decreasing the degree of change.

Since, with this variant, the entire air flow volume always passes through the HRU, it must have a correspondingly large size. The degree of temperature change depends on the outside air quantity, however. Therefore, the results shown in the following are applicable for all HRU types.

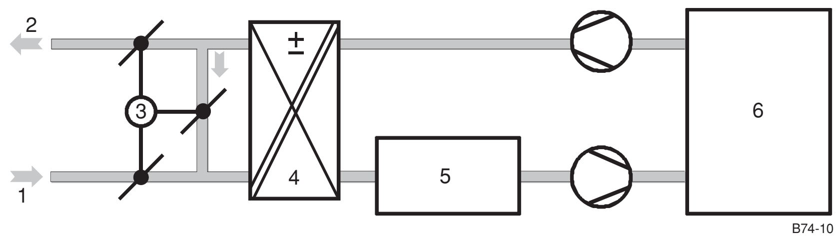


Fig. 4-13 Recirculated air mixing upstream of the HRU

1 Outside air
2 Exhaust air
3 Recirculated air mixing dampers
4 HRU
5 Supply air treatment
6 Room

Fig. 4-14 shows the progression of the overall degree of change of the combination as a function of the outside air quantity (a) for the HRU degrees of change $\Phi_{WR} = 0.5$ and $\Phi_{WR} = 0.7$ as well as the corresponding values for recirculated air mixing alone (Φ_M). It can be seen that the heat recovery at this outside air quantity is largely provided by the recirculated air mixing. The example plotted for a = 0.5 and $\Phi_{WR} = 0.5$ or 0.7 results in an overall degree of change of 0.67 or 0.77 for the combination and 0.5 for recirculated air mixing alone. Therefore, it would be economical to install an HRU in this case. The greater the outside air quantity the greater the proportion of heat recovery provided by the HRU. Therefore, the diagram clearly shows that, with an outside air quantity a > 70 %, addition of recirculated air mixing to the HRU is no longer worthwhile, and with an outside air quantity a < 30 %, addition of an HRU to the recirculated air mixing is no longer worthwhile.

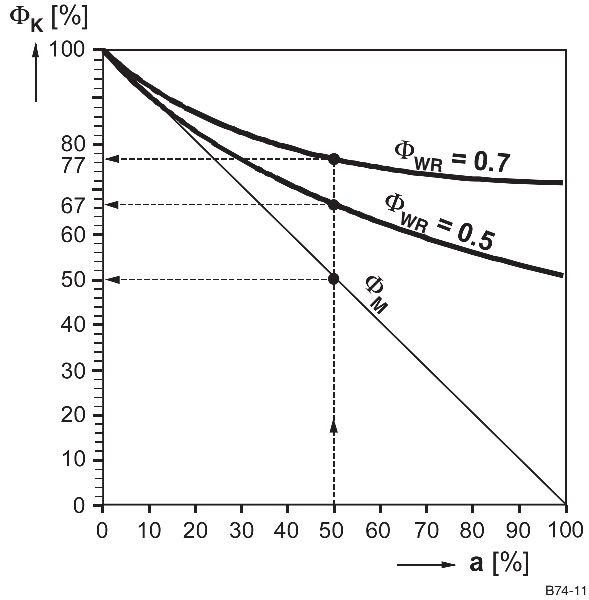


Fig. 4-14 Degrees of temperature change of the combination HRU and recirculated air mixing upstream of the HRU

a Proportion of outside air

Φ_K Degree of change of the combination

Φ_{WR} Degrees of change of the HRU

Φ_M Degree of change due to recirculated air mixing alone

Fig. 4-15 shows the progression of the mixed air temperature as a function of the outside air proportion with the degrees of temperature change $\Phi_{WR} = 0.5$ and $\Phi_{WR} = 0.7$ of the HRU as parameters. The example plotted for a = 50 % shows a mixed air temperature of 0 °C for $\Phi_{WR} = 0.5$ and –3 °C for $\Phi_{WR} = 0.7$.

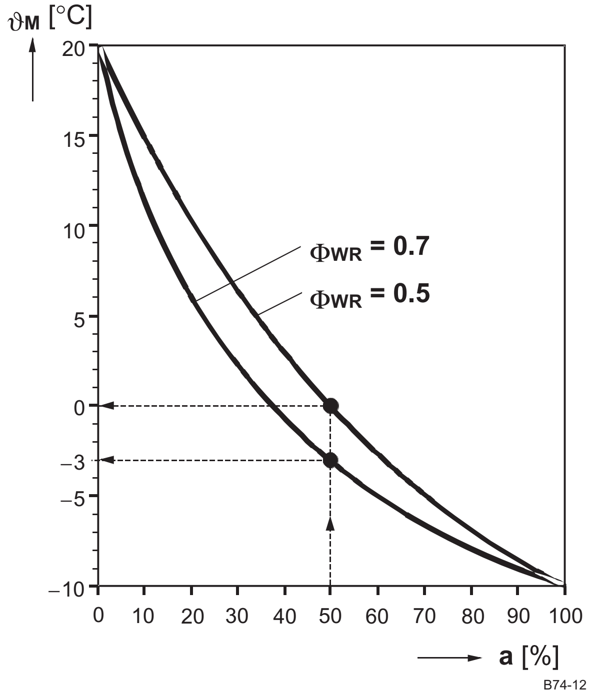


Fig. 4-15 Progression of the mixed air temperature as a function of the proportion of outside air (with an extract air temperature of 20 °C and an outside temperature of –10 °C)

Δ_M Mixed air temperature

a Proportion of outside air

Φ_{WR} Degrees of temperature change of the HRU

4.8.2 Recirculated air mixing downstream from the HRU

The configuration shown in Fig. 4-16 requires careful sizing of the mixing dampers so that this combination can be satisfactorily controlled.

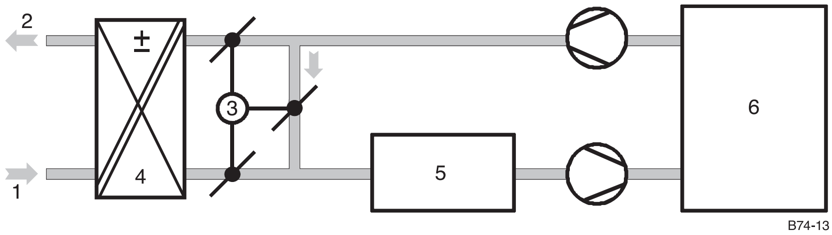


Fig. 4-16 Recirculated air mixing downstream from the HRU

1 Outside air
2 Exhaust air
3 Recirculated air mixing dampers
4 HRU
5 Supply air treatment
6 Room

If the following rough guidelines are followed, an acceptable control action with less than 10 % variation of the overall air flow can be expected:

- The bypass resistance should be twice that of the HRU
- All three dampers should be identically sized and operated simultaneously.
  The following damper types should be selected:
  - Parallel-blade dampers if the resistance of the HRU is < 50 % of the duct resistance. The damper resistance should be selected at approximately 20 % of the HRU resistance
  - Opposed-blade dampers if the resistance of the HRU is 50...100 % of the duct resistance. The damper resistance should be selected at approximately 10 % of the HRU resistance

Control

The HRU and the air dampers should be controlled sequentially. Devices requiring auxiliary energy are not brought online until after the recirculated air mixing:

Within the control range of the air dampers, the HRU is still switched off, and may even be bridged out. This means that only mixing of the two air flows occurs. Therefore, the operating characteristic of the damper system (outside air proportion as a function of the damper shaft rotation) must be linear. Additionally, the overall air flow should remain as constant as possible over the entire shaft rotation range.

Within the control range of the HRU, the mixing dampers remain in the maximum outside air proportion position. Therefore, the sizing and control of the HRU should be selected for these operating conditions.

Overall degree of change

The overall degree of change Φ_K of the combination can be calculated from the outside air quantity a and the degree of change of the HRU Φ_{WR}. The HRU degree of change is no longer constant in this case, but depends on the outside air proportion and the device type.

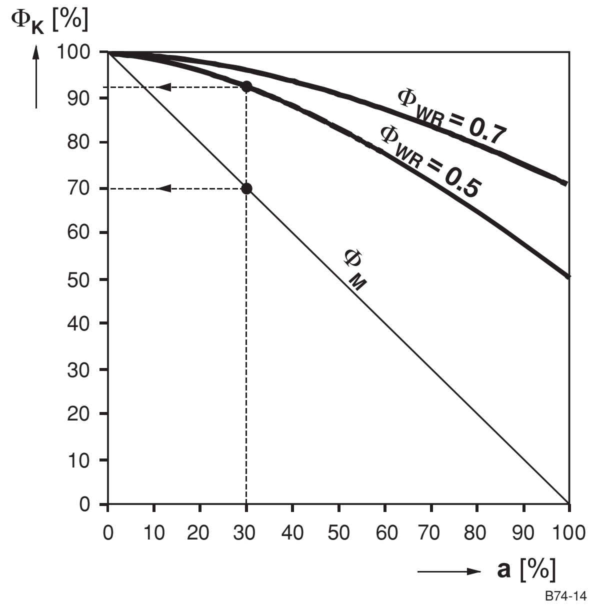


Fig. 4-17 Degree of change of the combination HRU with recirculated air mixing downstream from the HRU

a	Proportion of outside air quantity
Φ_K	Degree of change of the combination
Φ_{WR}	Degrees of change of the HRU
Φ_M	Degree of change due to recirculated air mixing alone

Fig. 4-17 shows the overall degree of change Φ_K as a function of the outside air proportion for the HRU degrees of change Φ_{WR} = 0.5 and Φ_{WR} = 0.7 for the combination of heat recovery with recirculated air mixing alone. The plotted example shows that, with an outside air proportion of only 30 %, the degree of change of 70 % with recirculated air mixing alone can be increased to over 90 % by combination with an HRU with Φ_{WR} = 0.5.

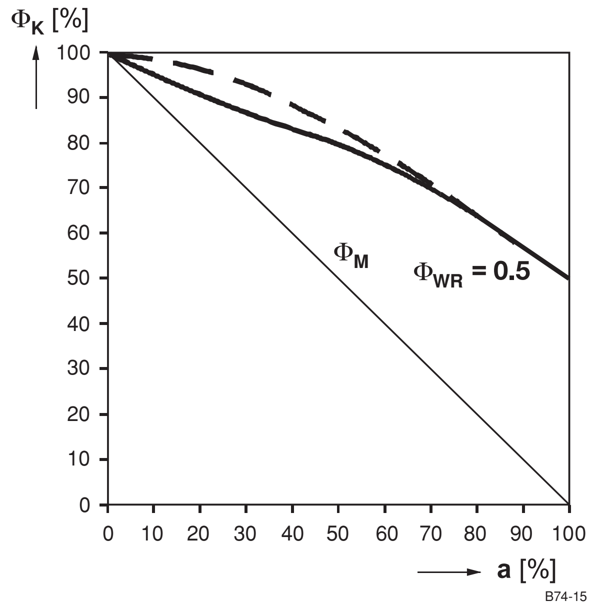


Fig. 4-18 Degrees of change of the combination closed circuit system and recirculated air mixing downstream from the HRU with (broken line) and without correction of the transfer medium circulation

a	Proportion of outside air quantity
Φ_K	Degree of change of the combination
Φ_M	Degree of change due to recirculated air mixing alone
Φ_{WR}	Degree of change of the closed circuit system

In the case of heat recovery with a closed circuit system, the degree of change can be further increased by correction of the circulation flow volume, because the progression of Φ_{WR} has a maximum. Fig. 4-18 shows the overall degree of change of the combination of a closed circuit system and recirculated air mixing for the HRU degree of change $\Phi_{WR} = 0.5$ as a function of the outside air proportion a with and without correction of the transfer medium circulation.

Therefore, the overall degree of change Φ_K can, at the same expense terms of equipment, be considerably increased with the combination of an HRU and downstream recirculated air mixing compared with recirculated air mixing upstream of the HRU. In comparing these two combinations, however, the anti-icing protection function of the combination with recirculated air mixing upstream of the HRU must also be considered.

5. Partial air conditioning system concepts

5.1 Design and fields of application

Partial air conditioning systems provide good results in a temperate climate, because the outside air humidity only rises above 65 % on a few days of each year. They are especially advantageous in systems with numerous internal heat sources Q_I, because dehumidification can be provided by cooling. In order to do so, however, the inlet temperature of the cooling medium must be at least 2 to 3 K below the dew point temperature of the supply air to be cooled.

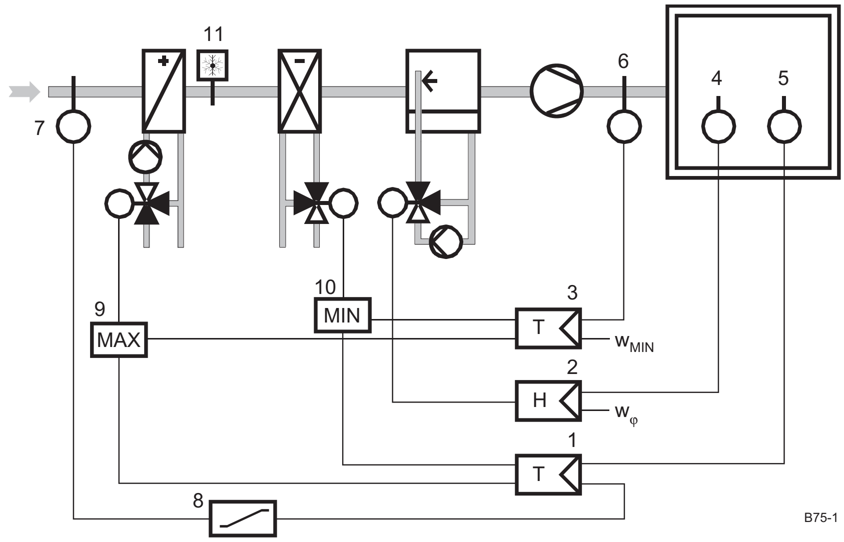


Fig. 5-1 Design of a partial air conditioning system
1 Room temperature controller
2 Room humidity controller
3 Supply air temperature low limit controller
4 Room humidity sensor (RH)
5 Room temperature sensor
6 Supply air temperature sensor
7 Outside temperature sensor
8 Setpoint shift controller
9 Maximum priority selection
10 Minimum priority selection
11 Frost protection thermostat

5.1.1 Temperature control

The room or supply air temperature controller controls the heating and cooling valve sequentially. The supply air temperature low limit controller (3) is used to prevent drafts. The room temperature setpoints can be adjusted for summer/winter operation via an outside temperature shift controller (8) or with dead zone control. Room/supply air temperature cascade control is recommended for larger spaces

5.1.2 Humidity control

A sensor acquires the room relative humidity, but the humidity controller acts on the humidifier valve only (no dehumidification). Various humidifier types can be used:
- Controllable air washer (humidification efficiency, see chapter 2)
- Controllable steam humidification (high limit control of supply air humidity required!)

In the case of the controllable air washer, the pump is switched off via an auxiliary switch when the humidifier valve is closed. Additionally, a low-cost air washer is sufficient, because the necessary room humidity can also be achieved with a medium humidification efficiency. It is a prerequisite, however, that the air washer is equipped with variable flow nozzles. Only partial dehumidification occurs when the temperature controller opens the cooling valve.

5.1.3 Air state changes on the psychrometric chart

Fig. 5-2 shows an example of the air state changes during heating operation, either with controllable saturated steam humidification or with a controllable air washer:

Assumptions:	Required room air state	20 °C / 40 % r.h.
	Outside air state	–5 °C / 60 % r.h.
	Required temperature difference	$\vartheta_{ZU} - \vartheta_R = 8$ K

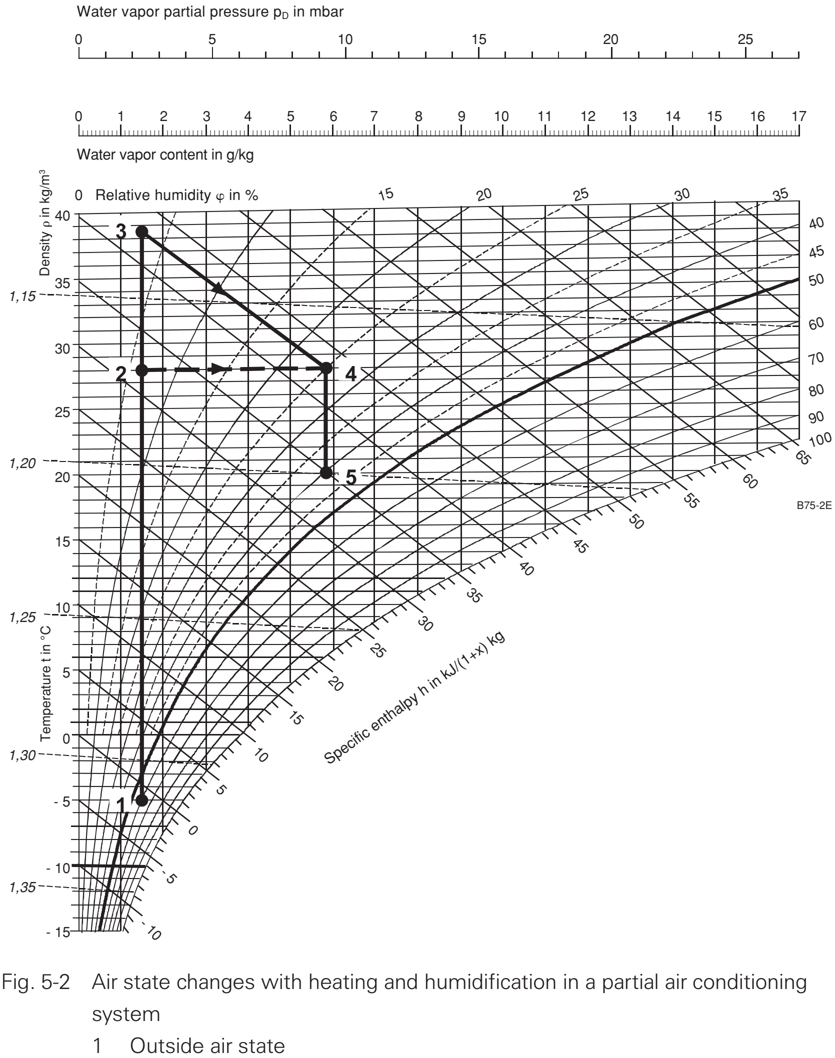


Fig. 5-2 Air state changes with heating and humidification in a partial air conditioning system

1 Outside air state
2 Required room air state
3 Steam humidifier inlet
4 Air washer inlet
5 Supply air inlet (heating, ϑ_R + 8 K)
3-5 Air state change with steam humidification
4-5 Air state change with controllable air washer

Fig. 5-3 shows an example of the air state changes during cooling operation with dehumidification via cooling:

Assumptions:	Required room air state	24 °C / approx. 50 % r.h.
	Outside air states (max.)	30 °C / 45 % r.h.
	Required temperature differential = $\vartheta_R - \vartheta_{ZU}$ = 7 K	
	Mean cooling surface temperature approx. 10 °C	

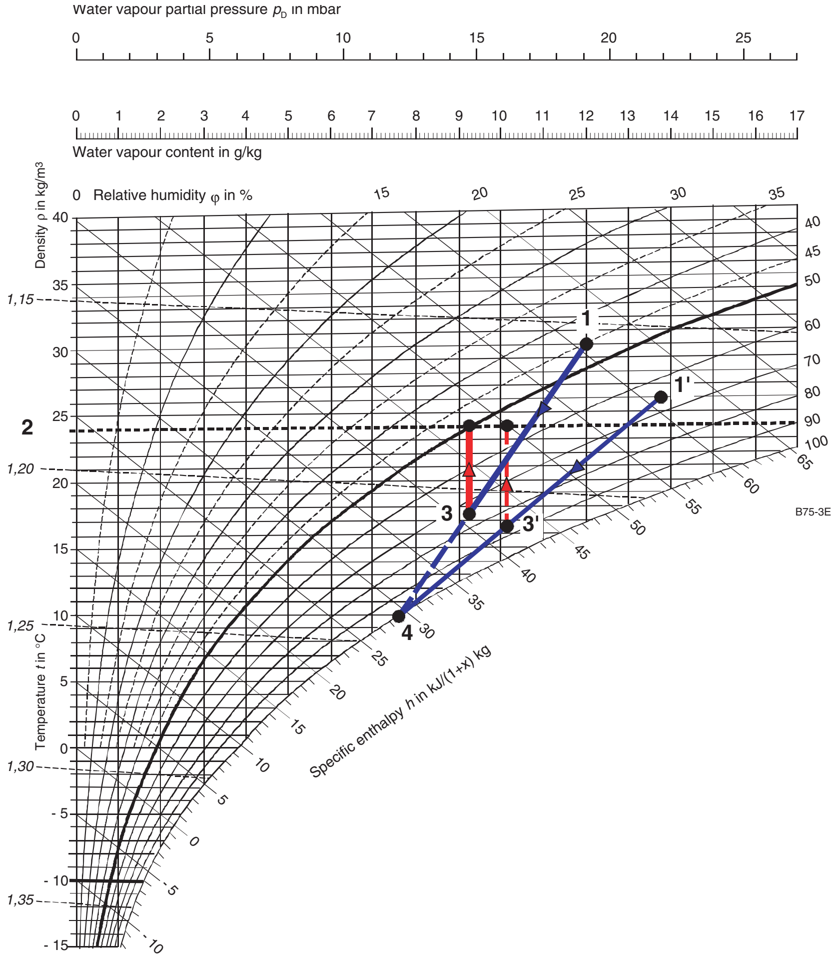


Fig. 5-3 Air state changes with cooling and humidification in a partial air conditioning system

1 (1') Outside air states (cooling coil inlet)

2 Room air temperature setpoint

3 (3') Supply air inlet (cooling coil outlet, ϑ_R – 7 K)

4 Mean cooling coil surface temperature

The plotted state changes apply to the assumed data only. If, for example, the outside air state (1) changes to state (1'), the room temperature controller commands the cooling coil outlet temperature to state (2') in order to maintain the room temperature at the current setpoint of 24 °C. Because the humidity controller has no effect on dehumidification in this case, the room relative humidity rises to approximately 55 % (state change shown with broken line).

5.2 Partial air conditioning systems for rooms with high internal humidity gain (swimming bath ventilation systems)

In special systems for rooms with high internal humidity gains (e.g. swimming pools) the air humidity of the conditioned space can, at low outside temperatures, be reduced by increasing the outside air proportion. This allows the maximum room humidity to be limited, but only as long as the water vapor content x of the outside air remains below that of the desired room air state. Otherwise, dehumidification without cooling is no longer possible.

5.2.1 Systems with modulating room temperature control and dehumidification via outside air mixing

In the partial air conditioning system (Fig. 5-4), the temperature and humidity of the conditioned space is modulated to constant limit setpoints. The room temperature sensor (1) acquires the controlled variable for the temperature controller (4). If it is below the selected setpoint, the controller's positioning signal opens the heating valve. However, if the room temperature is above the setpoint, the temperature controller cannot initiate cooling.

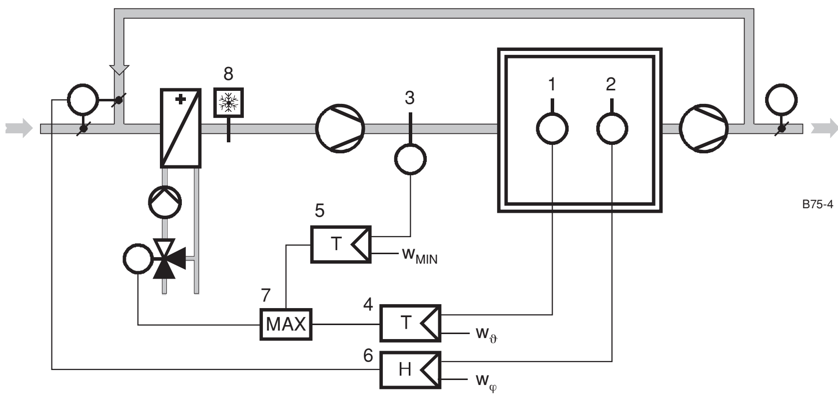


Fig. 5-4 Systems with modulating limit control of the room temperature and dehumidification via outside air mixing
1 Room temperature sensor
2 Room humidity sensor
3 Supply air temperature sensor
4 Temperature controller
5 Supply air temperature low limit controller
6 Humidity controller
7 Maximum priority selection
8 Frost protection thermostat

The temperature limiting controller (5) monitors the room inlet temperature via the supply air temperature sensor (3). If it falls below the selected limit setpoint, the controller acts on the temperature control loop via the maximum priority selection (7) and opens the heating valve. The humidity controller (6) compares the room humidity acquired by the sensor (2) against the selected limit setpoint. If it is above the setpoint, the controller opens the outside air damper via a positioning signal. Only as much outside air is supplied as is required to maintain the desired room humidity. If the room humidity is below the setpoint, however, no humidification occurs. If the water vapor content of the outside air rises above that of the room air, the room humidity limit setpoint can no longer be maintained.

5.2.2 Indoor air temperature and humidity control for a swimming pool, with compensation of the relative indoor humidity setpoint

The aim of this system strategy (Fig. 5-5) is not so much to maintain a comfortable level of indoor humidity as to prevent the formation of condensation on walls and window surfaces. This means that the relative indoor humidity must be reduced to a level at which the dew point temperature of the indoor air is always be lower than the lowest surface temperature of the enclosing surfaces ((Wi), i.e. walls, windows etc. of the space concerned. However, since this surface temperature varies as the outside temperature changes, it makes sense (to avoid wasting energy) to compensate the room humidity controller setpoint on the basis of the lowest wall or window surface temperature. There are special surface temperature sensors (window temperature sensors) available for this purpose.

The internal surface temperature calculation is based on the principles of heat transfer and thermal transmittance (see "Physical principles" in B01HV). The lowest internal wall or window surface temperature ϑ_{Wi} can be calculated simply with the following formula:

$$\vartheta_{Wi} = \vartheta_R - \Delta\vartheta_i$$

provided that the following information is known:

- Room temperature ϑ_R
- Outdoor temperature ϑ_{AU}
- Heat transfer coefficient, indoor air ⇨ wall α_i (α_i = 8 W/m²K for still air)
- Lowest k-value of external wall (this is normally the k-value of the windows)

The temperature differential $\Delta\vartheta_i$ to be calculated is determined using the following formula:

$$\Delta\vartheta_i = k - Wert * (\vartheta_R - \vartheta_A) * \frac{1}{\alpha_i} [K]$$

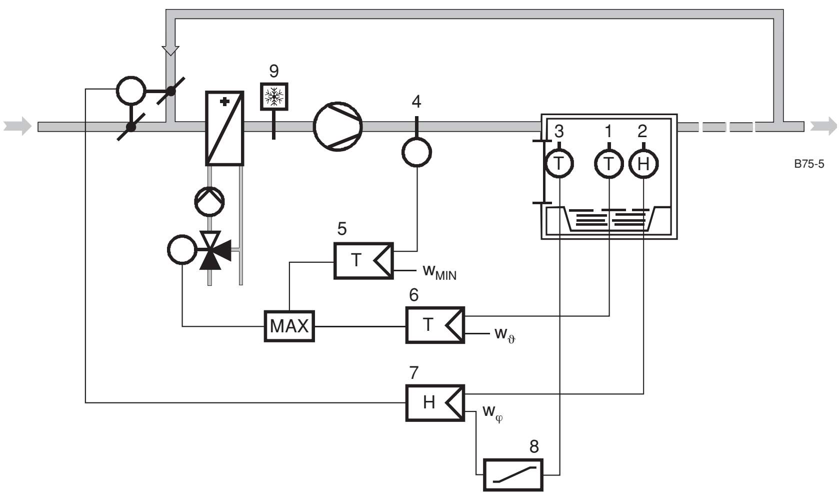


Fig. 5-5 Indoor air temperature and humidity control for a swimming pool, with setpoint compensation of the relative room humidity as a function of the window surface temperature

1 Room temperature sensor
2 Room humidity sensor
3 Surface temperature sensor
4 Supply air temperature sensor
5 Supply air temperature low limit controller
6 Temperature controller
7 Humidity controller
8 Setpoint compensation controller
9 Frost protection thermostat

If, as the result of a low outdoor temperature, the surface temperature of the external wall or the inside of the window falls to a level approaching the dew point of the indoor air (resulting from room temperature ϑ_R and relative room humidity φ_R), this is detected by the surface temperature sensor (3) and the room humidity setpoint is reduced in linear fashion via the shift controller (8) to a level low enough to prevent condensation on wall and window surfaces.

Example In a system, the relative indoor humidity setpoint is to be compensated as a function of the temperature of the internal window surface. The room temperature ϑ_R is to be controlled at 28 °C. The (glass) windows have a k-value of 2 W/m²K, and a value α_i of 8 W/m²K can be assumed for the internal heat transfer value.

The first task is to determine the maximum allowable indoor humidity setpoint when the outdoor temperature ϑ_A is at its lowest. This temperature depends on the location of the plant and can be seen in the plant planning documents. For the purposes of our example, the lowest temperature to be taken into consideration is –15 °C.

It is now possible to determine the surface temperature associated with this outdoor temperature. Under the conditions described above, this is calculated as follows (see further above for an explanation):

$$\Delta\vartheta_i = k-\text{Wert}\ (\vartheta_R - \vartheta_A) * \frac{1}{\alpha_i}$$

$$\Delta\vartheta_i = \frac{2W}{m^2K}\ (28-(-15))K * \frac{m^2K}{8W} = 10.75\ K$$

$$\Rightarrow \vartheta_{Wi} = \Delta_R - \Delta\vartheta_i = 28\ °C - 10.75\ K = 17.25\ °C$$

This information can now be used to determine the allowable indoor humidity setpoint from the h,x chart (cf. Fig. 5-9; 1-2-3-4). It will be apparent that the room humidity setpoint φ_R must be kept below 50 % rh to prevent the formation of condensation on the window surface.

Other operating points (e.g. at ϑ_A = +5 °C) can now be investigated and the associated permissible indoor humidity levels ascertained. It will then be possible to draw the applicable limit curve as a function of the outside or surface temperature (cf. Fig. 5-7, broken line). Hence, to avoid the formation of condensation on walls and windows, the indoor humidity must be maintained below 70 % at a temperature of +5 °C, and below 50 % at a temperature of –15 °C. In practice, the actual setpoint compensation value is set slightly below the limit curve (cf. Fig. 5-11, w_φ).

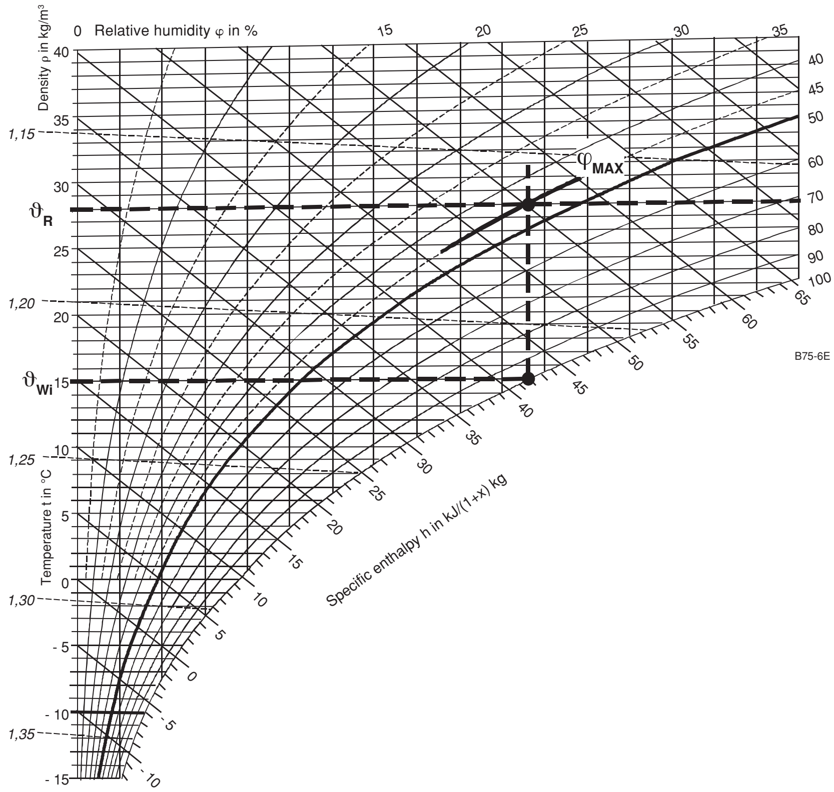


Fig. 5-6 Using the psychrometric chart to determine the maximum permissible room relative humidity based on the room temperature and the lowest wall or window surface temperature

ϑ_{Wi} Lowest wall or window surface temperature

ϑ_{R} Room temperature

φ_{MAX} Maximum relative room air humidity (condensation limit)

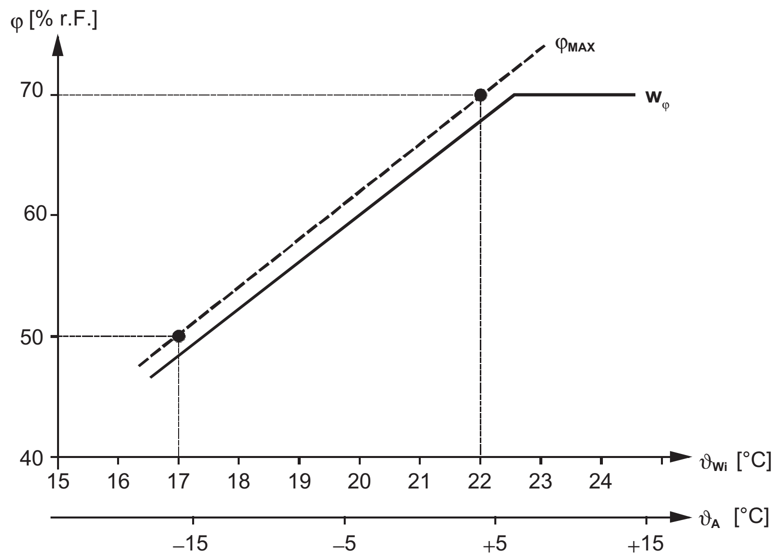


Fig. 5-7 Linear set point compensation of room relative humidity based on the lowest wall or window inner surface temperature

ϑ_{Wi} Lowest wall or window inner surface temperature

ϑ_{AU} Outside temperature

ϑ_{MAX} Maximum relative room air humidity (condensation limit)

w_{φ} Set point compensation for prevention of condensation

5.2.3 Indoor temperature and humidity control for a swimming pool, with relative indoor humidity setpoint compensation and heat recovery

The partial air conditioning system shown in Fig. 5-8 is a suitable, comfortable concept for relatively large swimming pools. The slow control behavior of a large space can be easily managed with room/supply air cascade control. The room temperature controller (6, P-controller) compensates the setpoint of the supply air controller (5, PI controller) between a minimum and a maximum value, which also includes the low and high limit control of the supply air temperature.

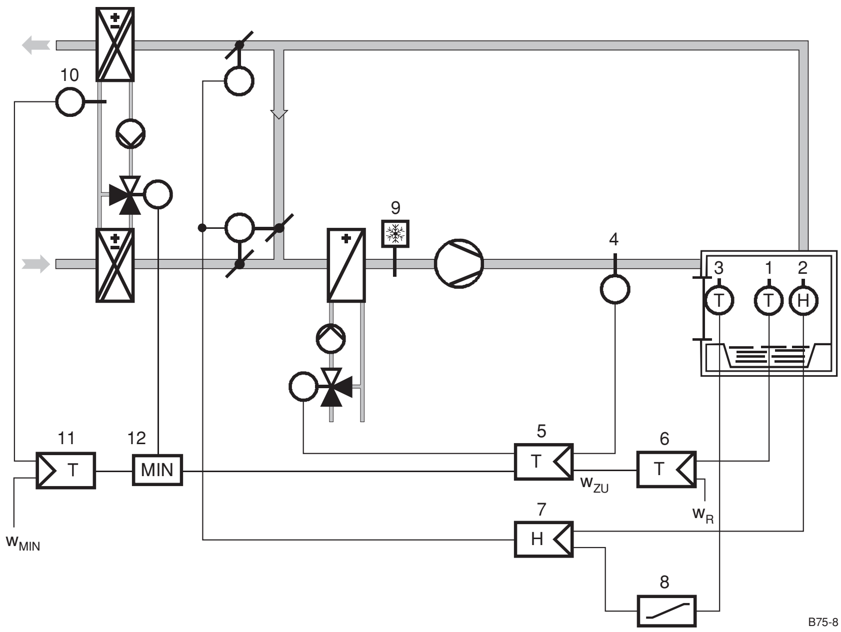


Fig. 5-8 Indoor temperature and humidity control for a swimming pool, with setpoint compensation of the relative indoor humidity based on window surface temperature

1 Room temperature sensor
2 Room humidity sensor
3 Surface temperature sensor
4 Supply air temperature sensor
5 Supply air temperature controller
6 Room temperature controller
7 Humidity controller
8 Setpoint shift controller
9 Frost protection thermostat
10 Anti-icing protection sensor
11 Anti-icing protection controller
12 Minimum priority selection

The relative indoor humidity control strategy is the same as that described in 5.2.2, so it will not be described again here.

In a relatively large and frequently-used swimming pool, where a considerable quantity of outside air must be continually mixed in order to reduce the humidity of the room air, it is worthwhile installing a closed circuit HRU (extract air duct is physically separated from the supply air treatment plant). The necessary anti-icing protection control (with sensor (10) and controller (11) can act on the positioning signal of the HRU sequence of the supply air temperature controller (5) via a minimum priority selection (12).

5.3 Partial air conditioning system with heat recovery – heating – cooling

Fig. 5-9 shows the schematic of this system with a suitable control concept.

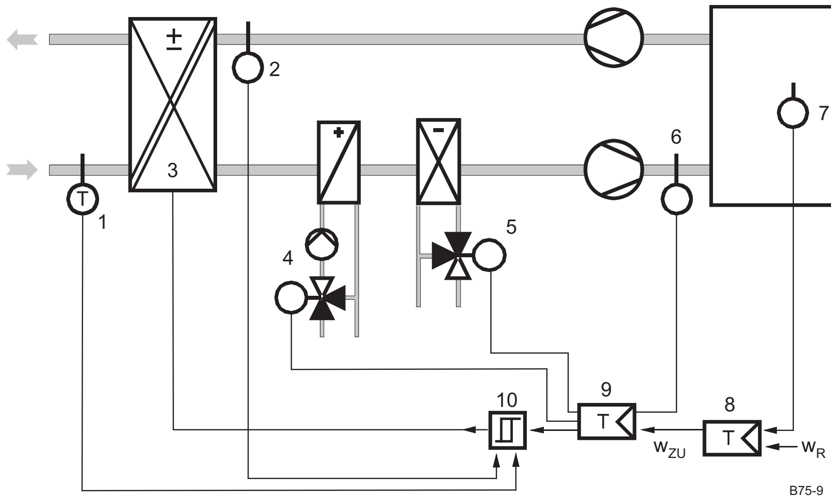


Fig. 5-9 Schematic of a partial air conditioning system with heat recovery, heating, cooling

1 Outdoor sensor
2 Extract air temperature sensor
3 HRU
4 Motorized heating valve
5 Motorized cooling valve
6 Supply air temperature sensor
7 Room temperature sensor
8 Room temperature controller
9 Supply air temperature controller
10 Two-position changeover switch (HRU max./min.)

The heat recovery system used here will change the heat flow direction depending on whether the outside air temperature is lower or higher than that of the extract air. If the outside temperature is lower than the extract air temperature, the temperature at the outlet of a thermal wheel heat exchanger rises if the rotor speed is increased. If, however, the outside air temperature is higher than that of the extract temperature, its outlet temperature at the rotor will fall with increasing rotor speed.

These temperature differences, which give rise to an inversion of the action of the control loop, are detected by the outside air and extract air sensors and taken into account by the HRU changeover function: as soon as the outside air temperature (1 is higher than the extract air temperature (2, the heat recovery is commanded to maximum operation, regardless of the current control signal (Fig. 5-10). This applies to all HRUs in which the heat flow direction can change.

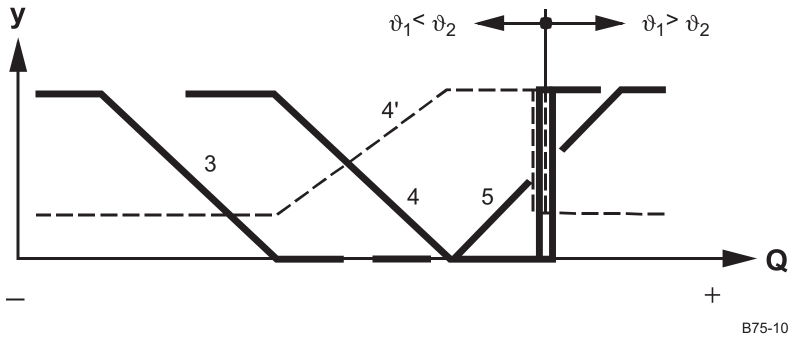


Fig. 5-10 Stroke/load diagram for the schematic in Fig. 5-9

y	Stroke (manipulated variable)
Q	Load (– = heating load, + = cooling load)
ϑ_1	outside air temperature
ϑ_2	extract air temperature
3	Heating valve
4’	Air damper actuator or outside air damper (instead of HRU)
4	HRU manipulated variable
5	Cooling valve

Control concept

The controller must command three devices (HRU, air heating coil, air cooling coil) with different static and dynamic properties. The controller output effect X_h (static control characteristic) of the HRU is determined by the temperature or enthalpy difference between the extract air and outside air, so it varies greatly. The controller output effect of the HRU is also usually much smaller than that of the air heating coil. If the HRU is commanded by the controller, the control loop is hyperstable, i.e. too slow. This problem is alleviated to some degree by the selection of room/supply air cascade control, because the control dynamics are considerably improved by the auxiliary control loop.

6. Concept of full air conditioning plant with heat recovery

6.1 General

Definition

An air treatment system is referred to as a full air conditioning plant if both the air temperature and the air humidity are controlled to selected setpoints.

A ventilation or air conditioning system with a heat recovery system should be operable in such a way that the energy required for supply air treatment is minimized. The heat exchange output of the heat recovery system that is required at a given time is determined by the following variables:

- Temperature and relative humidity of the outside air, extract air and supply air
- Humidification system type (steam humidifier or air washer)

Why control the HRU?

In certain operating cases with heat recovery, the reclaimed heat can exceed the current requirement for supply air treatment. This can readily occur in case of a large external heat gain in the conditioned space. Systematic investigations have shown that HRUs must be operated with reduced heat exchange output during 20 to 40 % of their operating time. If the heat exchange output is not reduced in these cases, the reclaimed heat must be cooled away again with additional energy expenditure in the supply air treatment system. Therefore, the heat exchange output of an HRU must be adapted to requirements by open or closed loop control.

Specific problems of heat recovery control

If the heat recovery system is to be operated with open loop control, the energy demand must be measured in locations where the measured variables cannot be influenced by the HRU. However, this condition is only met in the case of the outside air. However, the outside air temperature or enthalpy provides little indication of the system's energy requirements, because the external heat sources in the conditioned space and in the supply air are not included. Therefore, energy-optimized open loop control of the heat recovery system is not possible over the full operating range. This raises the question whether the heat recovery system should be equipped with its own control loop or included as a sequence in the overall air treatment control.

If the heat recovery system were operated with its own control loop, taking the supply air temperature or enthalpy as the reference variable and the air temperature or enthalpy at the outlet of the heat recovery system as the controlled variable, the heat recovery control would be highly intermeshed with the supply air or room temperature control. Therefore, stable control operation would not be possible.

Since, in the case of reduced heat recovery operation, the energy for supply air heating is supplied by external heat sources and heat recovery only, the integration of the heat recovery system as a sequence of the room or supply air control suggests itself. This sequence control is proposed for the major air treatment system types in the following. The functional design is mainly based on static operating states.

When planning a full air-conditioning system with heat recovery, the question arises as to whether the temperature difference ($\Delta\vartheta$) or the enthalpy difference (Δh) between extract air and outside air should be selected as the changeover criterion for the change of heat flow direction. This decision depends, on the one hand, on the type of heat recovery (with or without humidity transfer) and on the humidification method (steam humidification or air washer) on the other.

Fig. 6-1 shows a decision matrix for the selection of the correct reference variable ($\Delta\vartheta$ or Δh) for the heat recovery changeover function. The matrix is intended as a rough decision-making guide, and it applies to normal outside air states in temperate climatic zones. It is recommended to verify the actual state changes using the psychrometric chart in every case. Changeover based on Δh only makes sense if an air washer is used in conjunction with a recuperative heat recovery unit or with recirculated air dampers, because the change of state in the h,x chart follows the line of enthalpy (adiabatic).

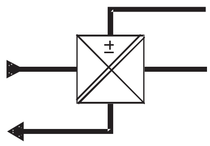
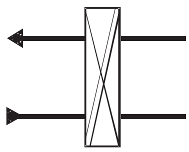
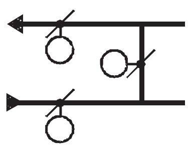
Heat reclaim	**Humidifier type**	**Changeover function based on:** $\Delta\vartheta$	Δh
RECUPERATIVE 	None	●	
	Steam humidifier	●	
	Air washer	●	
REGENERATIVE 	None	●	
	Steam humidifier	●	
	Air washer		●
MIXING DAMPERS 	None	●	
	Steam humidifier	●	
	Air washer		●

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Fig. 6-1 Decision matrix for heat recovery changeover based on $\Delta\vartheta$ or Δh

6.2 Full air conditioning system with recuperative heat recovery – cooling – heating – humidification (steam) – dehumidification

Control concept

Fig. 6-2 shows a schematic of a full air conditioning system with recuperative heat recovery and steam humidification. Room/supply air temperature cascade control is selected as a suitable control concept in this case. An additional control loop has the task of maintaining the room humidity setpoint.

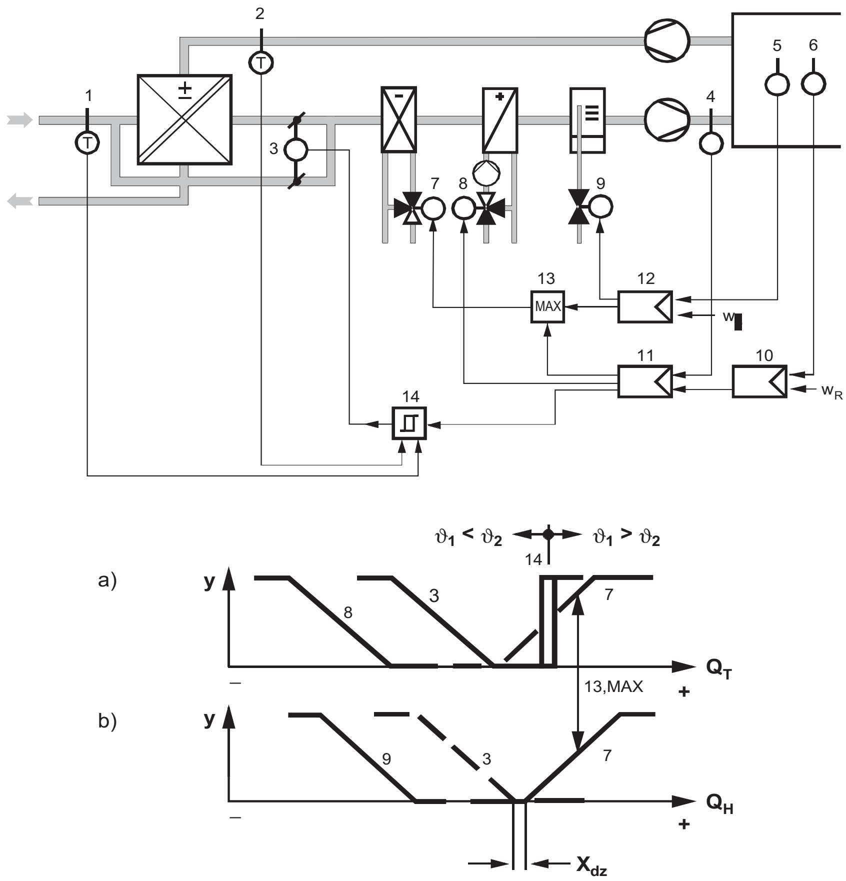


Fig. 6-2 Schematic of a full air conditioning system with recuperative heat recovery, cooling, heating, steam humidification and dehumidification

1 Outside air temperature sensor
2 Extract air temperature sensor
3 HRU control dampers
4 Supply air temperature sensor
5 Room humidity sensor
6 Room temperature sensor
7 Cooling valve
8 Heating valve
9 Steam humidifier valve
10 Room temperature controller (primary controller)
11 Supply air temperature controller (auxiliary cascade controller)
12 Room humidity controller
13 Maximum priority selection: Cooling/dehumidification
14 Heat recovery max./min. changeover function
ϑ_1 Outside air temperature
ϑ_2 Exhaust air temperature
a) Temperature control stroke/load diagram
Q_T – = Heating load, + = Cooling load
b) Humidity control stroke/load diagram
Q_H – = Humidification load, + = Dehumidification load

The humidification function is accomplished via a steam valve (9), and the dehumidification function acts, together with the cooling function of the temperature controller, via a priority selection (13) on the cooling valve (7). The room temperature and relative humidity are controlled to constant values. If the room humidity needs only to be kept within specified limits, this can be achieved by adding a dead zone x_{dz}, as shown on the stroke/load diagram. Since neither humidification nor dehumidification is required within the dead zone, this operating mode offers increased economy and is, therefore, recommendable. If dehumidification is to be completely dispensed with for reasons of cost, this can be achieved by omitting the HRU and cooling sequence in the humidity control loop. The acceptability of such a measure must checked with respect to the meteorological data.

6.3 Full air-conditioning system with recirculated air mixing – cooling – heating – humidification (continuous, adiabatic) – dehumidification

This system is similar in structure to the one just described. However, outside air / exhaust air / recirculated air dampers are used instead of a heat recovery unit, and enthalpy sensors are installed in the outside air and extract air duct for the damper changeover (cf. Fig. 6-3).

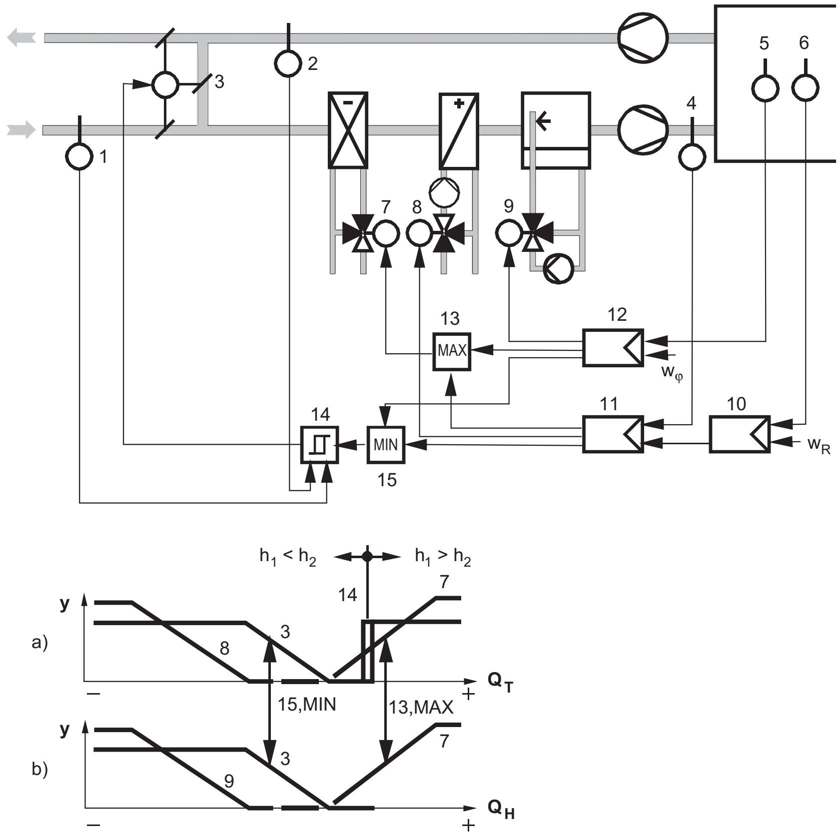


Fig. 6-3 Schematic of a full air conditioning system with heat recovery dampers, cooling, heating, adiabatic humidification and dehumidification

1 Outside air enthalpy sensor
2 Exhaust air enthalpy sensor
3 HRU control dampers
4 Supply air temperature sensor
5 Room humidity sensor
6 Room temperature sensor
7 Cooling valve
8 Heating valve
9 Air washer valve
10 Room temperature controller (primary controller)
11 Supply air temperature controller (auxiliary cascade controller)
12 Room humidity controller
13 Maximum priority selection: Cooling/dehumidification
14 Heat reclaim max./min. switchover function
15 Minimum priority selection heat reclaim dampers: Heating/humidification
h_1 Outside air enthalpy
h_2 Exhaust air enthalpy
a) Temperature control stroke/load diagram
Q_T – = Heating load, + = Cooling load
b) Humidity control stroke/load diagram
Q_H – = Humidification load, + = Dehumidification load

The proposed control system also operates with optimal energy usage in this case. Unlike in the previous examples, the heat recovery change-over is not governed by a temperature comparison but by an enthalpy comparison between outside air and extract air.

If the relative humidity is only specified between two limits, it can also be achieved in this case by insertion of a dead zone (with PI control) between the humidification and dehumidification functions. If dehumidification is dispensed with for reasons of cost, the corresponding sequence can be omitted in the humidity control loop. However, the heat recovery sequence is retained, because it can be used for dehumidification at times (heating operation) in case of a large humidity gain in the conditioned space.

In the case of air conditioning systems that frequently operate with dehumidification, the air humidity is removed using a chiller whose evaporator and condenser are installed in the supply air duct.

6.4 Full air conditioning system with recuperative heat recovery – cooling – heating – humidification (continuous, adiabatic) – dehumidification

Fig. 6-4 shows the schematic of a system of this type. If this system is to operate with optimum energy usage, considerably more complicated operating equipment is required than for the previous examples. A control concept like the one shown in Fig. 6-2 provides energy usage that is satisfactorily close to the optimum. The heat recovery changeover function is based on the temperature difference between outside air and extract air.

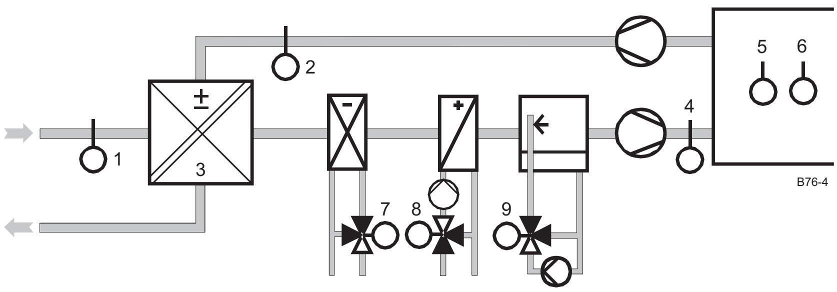


Fig. 6-4 Schematic of a full air conditioning system with recuperative heat recovery, cooling, heating, adiabatic humidification and dehumidification (control as in Fig. 6-2)

1 Outdoor sensor
2 Extract air temperature sensor
3 Recuperative HRU
4 Supply air temperature sensor
5 Room temperature sensor
6 Room humidity sensor
7 Motorized cooling valve
8 Motorized heating valve
9 Air washer valve

6.5 Full air conditioning system with regenerative heat recovery (humidity transfer) – cooling – heating – humidification (steam) – dehumidification

Highly complex control would also have to be selected for the system example as per the schematic shown in Fig. 6-5 in order to operate with optimum energy usage. However, satisfactory results can also be achieved in this case with the control concept as per Fig. 6-3 (with minimum priority selection 15).

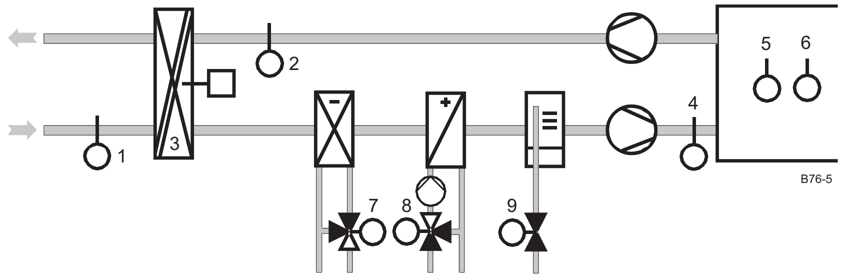


Fig. 6-5 Schematic of a full air conditioning system with regenerative heat recovery (humidity transfer), cooling, heating, steam humidification and dehumidification (control as in Fig. 6-2)
1 Outdoor sensor
2 Extract air temperature sensor
3 Regenerative HRU
4 Supply air temperature sensor
5 Room temperature sensor
6 Room humidity sensor
7 Motorized cooling valve
8 Motorized heating valve
9 Steam humidifier valve

6.6 Comparison of the various system concepts with regard to energy consumption

The exact energy consumption of the various air treatment system types can only be determined with a major calculation effort. However, a qualitative comparison regarding energy expenditure can be readily made by observing the air treatment processes on the psychrometric chart. The system types described above will be briefly evaluated and compared in the following:

- Steam humidification with recuperative heat recovery (Fig. 6-2): This system type will use the most energy, because natural cooling due to humidification and humidity transfer via the HRU are not present
- Continuous adiabatic air humidification and regenerative heat recovery or dampers (Fig. 6-3): This system type is bound to use the least energy for air treatment, because the adiabatic humidification provides cooling without additional energy expenditure, and humidity is transferred via the heat recovery (humidification and dehumidification)
- Continuous adiabatic humidification and recuperative heat recovery (Fig. 6-4):
  In this system type, no humidity transfer via the heat reclaim occurs. Therefore, increased energy expenditure is required for the necessary additional humidification and dehumidification
- Steam humidification with regenerative heat recovery or damper (Fig. 6-5):
  Natural cooling via the humidifier does not occur. This quantity of cooling must be provided by the chiller. Therefore, the energy consumption is higher than in a corresponding system with an air washer

6.7 hx-directed control (Economiser tx2)

Little has changed in the past few decades in relation to the strategy for control of air conditioning systems. Typical features are the heat recovery systems described above based on enthalpy, or separate temperature and humidity control loops each with their own dead zones, and no mutual coordination. There is no doubt that there is untapped potential here for optimization.

With hx-directed control (Economiser tx2) this potential is exploited simply by making consistent good use of the comfort limits and by mutual coordination of the setpoint, process control and energy recovery.

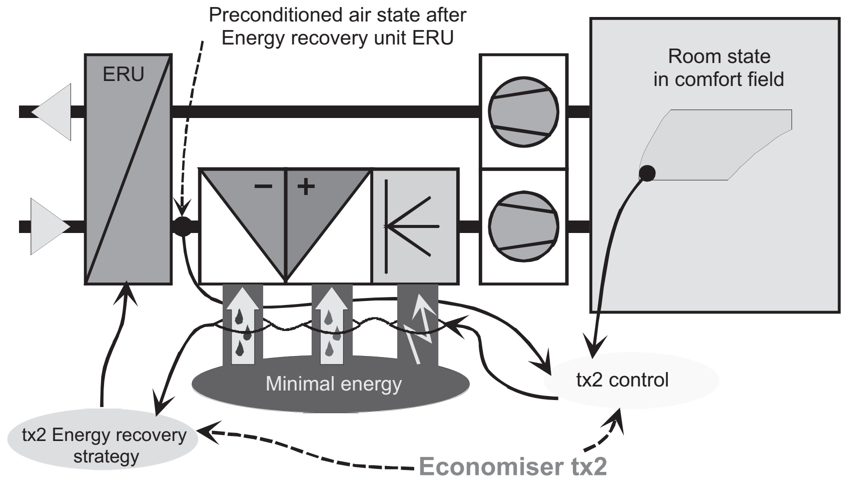


Fig. 6-6 Principle of the Economiser tx2

Objective Let us assume as our starting point a full air conditioning system ready for operation, with heat recovery (cf. e.g. Fig. 6-2). The aim of this air conditioning system is to create **adequate comfort conditions** (indoor conditions suitable for occupancy) **at the lowest possible operating costs**. The operating costs must not be minimized at the expense of comfort.

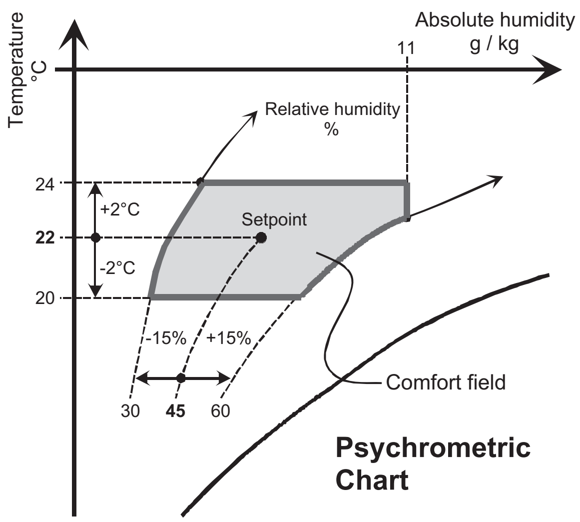


Fig. 6-7 Comfort field as the basis for h-x directed control

The Economiser tx2 (t = temperature, x = absolute humidity, 2 = control of 2 variables) finds the best possible room setpoint at the limits of or within the comfort field, (Fig. 6-7) and makes optimum use of energy recovery. This is achieved by tx2 control (see 6.7.1) and the tx2 energy recovery strategy (see 6.7.2).

In order to make optimization at all useful, we need to know the relative costs of the heating, cooling, humidification and dehumidification processes (see 6.7.2, Weighting the processes), and their physical properties. Fig. 6-8 provides an overview of the process properties of which the Economiser tx2 takes account.

Process	General description	Example
Heating	Absolute humidity remains constant	Plate heat exchanger
Cooling	Absolute humidity remains constant	Plate heat exchanger
Humidifcation	Temperature-constant enthalpy-constant	Steam humidifier Air washer
Dehumidification	Drop in temperature and humidity	Plate heat exchanger
Energy recovery	Partial recovery of sensible and latent heat	Regenerative systems (thermal wheel)
	Partial recovery of sensible heat	Recuperative systems (plate heat exchangers, closed-circuit heat exchangers)
	Complete recovery of sensible and latent heat	Recirculated air dampers

Fig. 6-8 Process properties
Energy recovery and humidification options can be freely combined

6.7.1 tx2-control

Cascade control

The dynamic controller consits of a cascade controller for the temperature and humidity control loops.

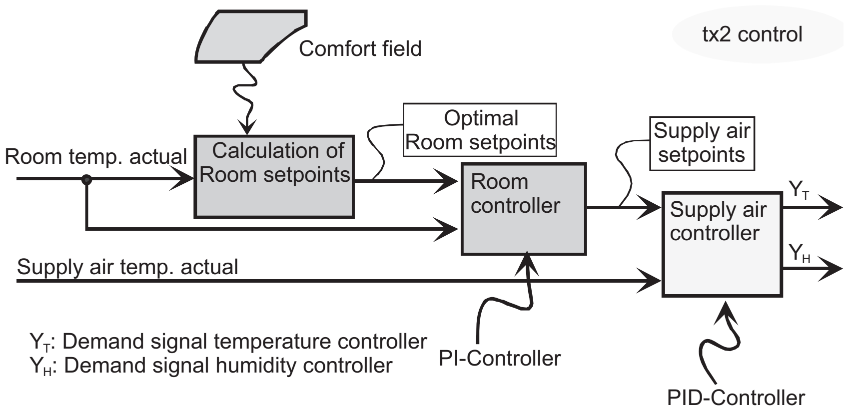


Fig. 6-9 Cascade control
tx2 control is based on separate cascade controllers for the temperature and absolute humidity respectively

The measured variables are temperature and relative humidity, but the reference variables are temperature and absolute humidity.

The absolute humidity is calculated from the temperature and the relative humidity. With this calculation, the two reference variables are separated, which results in a better dynamic response. The cascade incorporates three important components:

- Determination of the setpoint at the limits of or within the comfort field
- Room controller
- Supply air controller

Determining the setpoint

Based on the condition of the air, pretreated by the energy recovery system, and the measured value in the room, the optimum room setpoint is defined at the limits of or within the comfort field.

The condition of the pretreated air, combined with the supply air setpoint, results in a heating, cooling, humidification or dehumidification demand, or a combination of these.

- In the case of the combinations heating/humidification, cooling/humidification and dehumidification/reheating, the relevant setpoint is identical to the best vertex in energy terms of the comfort field.
- In the case of cooling-only, the setpoint is determined by projecting the measured value along the upper border of the comfort field.
- With heating-only, the setpoint is projected allowing the lower border of the comfort field. The area where the limit is defined by relative humidity is a special case. Here, the temperature setpoint is shifted along the line of relative humidity. This leads to increased heating, balanced by the fact that no cooling is required within this field. The result is an overall reduction in costs, since heating is normally cheaper than cooling.
- In the case of humidification only, the measured humidity is projected along the line of relative humidity. This leads to increased humidification, balanced by the fact that no cooling is required within this field. The result is an overall reduction in costs, since humidification is normally cheaper than cooling.

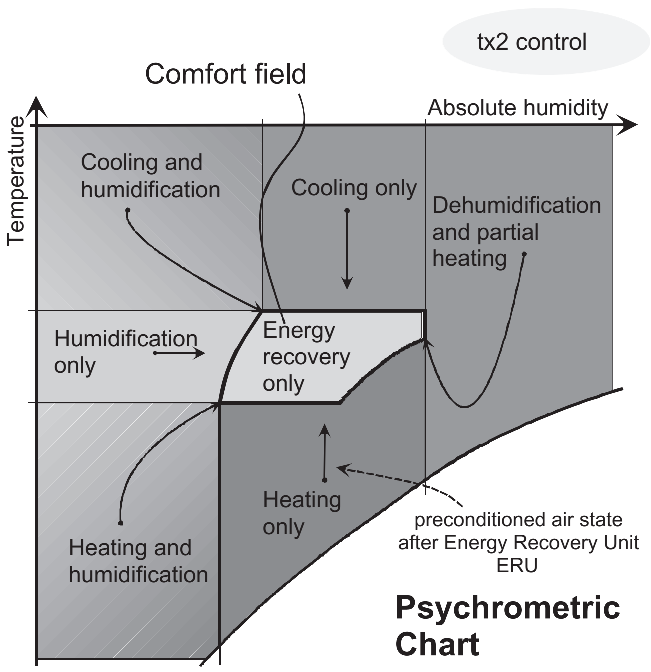


Fig. 6-10 Possible changes of state in the hx chart, to reach the comfort field

Room controller The room controller calculates the supply air setpoint, based on the control differential in the room. The room controller incorporates two PI-controllers, one for the temperature control loop and one for the humidity control loop.

Supply air controller The supply air controller calculates demand signals Y_T and Y_H based on the control differential on the supply air side. It consists of PID controllers for the temperature and humidity control loops. The demand signals are used to position the control elements in the air conditioning plant and to control the heat recovery system.

6.7.2 tx2 energy recovery strategy

Strategy The energy recovery system is controlled in such a way as to minimize the sum of the weighted demand signals for the heating, cooling, humidification and dehumidification processes. The energy recovery strategy is based on a modeled approach to the tx diagram. Each process is assigned a vector at the tx-level and a weighting (see Fig. 6-11). The vectors reflect the theoretical effect of each process. However, apart from the heat recovery vector, only two other vectors may ever be in operation at any one time.

The supply air setpoint is entered in the middle of the diagram. The measured room value and the outside air value are also entered. Then, starting from the measured outside air value, the energy recovery vector is drawn in, depending on the degree of energy recovery from temperature and humidity.

Now, the objective is to find the point on the energy recovery vector, which minimizes the sum of the theoretical effects of the two active processes.

Example The outside air is cooler than the room air. In the event of a high cooling demand, the heat recovery system should also be used to treat the cooler air, thereby increasing the proportion of outside air. This action reduces the amount of cooling energy required. If there is a need for simultaneous humidification, this adds a further dimension (see the algorithm).

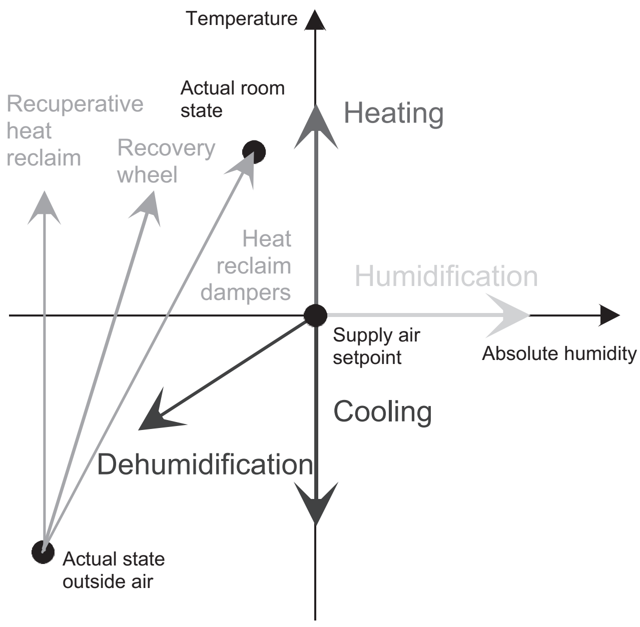


Fig. 6-11 tx diagram
To achieve optimum separation of the processes, the Economiser tx2 works with the variables temperature and absolute humidity.

Weighting the processes

The energy recovery system optimizes energy consumption by preconditioning the outside air. For this purpose, the demand signals from the supply air controller (6.7.1) are weighted with the specific costs of the associated processes (Fig. 6-12) and the energy recovery system is controlled by the method described further below.

Process	**Specific costs**	**Relative weighting**
Heating	0.09 €	1
Cooling	0.16 €	2
Humidification	0.08 €	1
Dehumidification	0.23 €	3

Fig. 6-12 Example showing specific costs and relative weighting
The specific costs are in € and relate to a temperature differential of 1 K and a period of operation of 1 hour
The relative weighting values are used to optimize the energy recovery system

Algorithm

The process described in the patent application[1] for calculating the control signal for energy recovery is described below. The steps can be followed by reference to the numbers in Fig. 6-13.

1. Enter supply setpoint and measured outside air and room air values on the tx diagram.
   The energy recovery vector is divided into room and outside air values coupled with the type of energy recovery used. The aim is to reach the supply air setpoint. In practice however, only points along the energy recovery vector can be reached with the energy recovery system.
2. Enter the weighted demand signals as vectors (in the example, these are the vectors for heating and dehumidification).
   Weighted demand vectors show the required response from the controller. The heating vector calls for an increase in temperature hence points upward on thee temperature axis. Dehumidification calls for a reduction in absolute humidity. This, however, can only be achieved by a reduction in the cooling unit to below the dew point temperature. This is why cooling is also required, and explains why the dehumidification vector includes a temperature component.
3. Project the supply air setpoint and the weighted demand vectors onto the energy recovery vector.
   By projecting the weighted demand vectors, more detailed information on the position of the energy recovery vector is possible. If, for example, the absolute humidity of the room air is the same as that of the outside air, there is no possibility of humidity recovery. In the event of an active humidity demand, the projection produces the zero vector. In this case, therefore, the weighted demand vector has no effect, and only the temperature demand vector determines the control signal for energy recovery. The projected supply air setpoint provides a starting point for addition on the energy recovery vector.
4. Add the projected demand vectors starting from the projected supply air setpoint.
   The projections of the demand vectors should be added together. Depending on the demand and the position of the energy recovery vector, the projected vectors will point in the same or the opposite direction, also indicating whether or not the same response is required from energy recovery system. If they do not point in the same direction, the weighting and the current demand will determine the type of energy recovery. In Fig. 6-13 the dehumidification vector determines the actual energy recovery effect, i.e. the projection is significantly higher than that of the heating vector.

[1] European patent application No. 97 100,822, published July 1997

5. Calculate the control signal for the energy recovery system by interpolating the point resulting from Step 4 between the room air (max. energy recovery e.g. 20 %) and the outside air (min. energy recovery 100 %).
   The result of Step 4 is a point on the energy recovery vector or in the extension of the energy recovery vector. If the point is on the vector, then the energy recovery control signal is determined by interpolation between the two end values. In this process, the room air is assigned a value of e.g. 20 % (maximum energy recovery) and the outside air 100 %% (minimum energy recovery). If the point is on the extension of the vector, the value used is that of whichever condition of the air is closer.

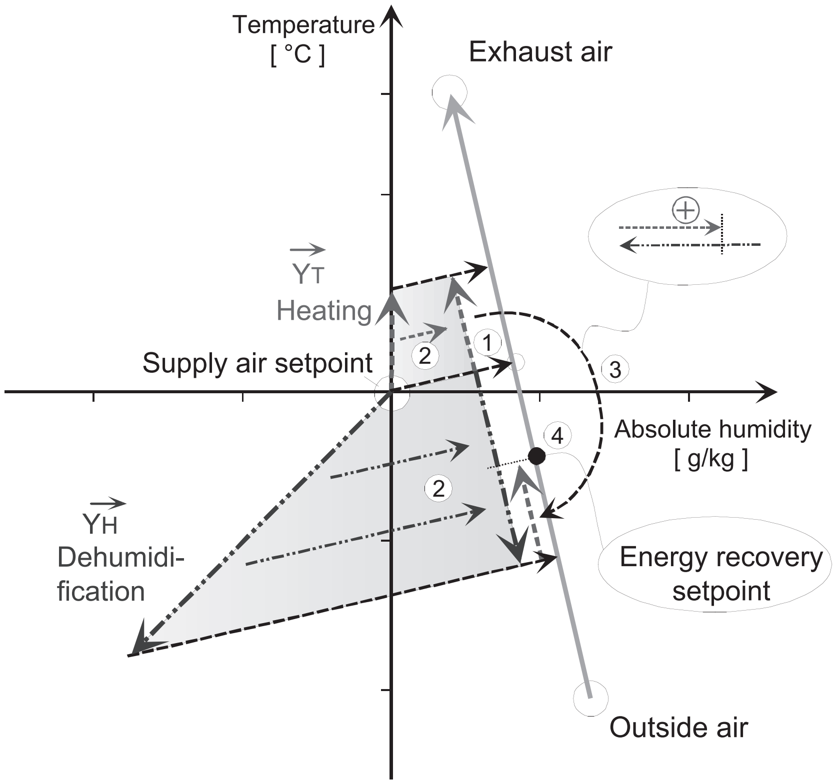


Fig. 6-13 Energy recovery control

--- Weighted heating demand signal
-··- Weighted humidification demand signal
— Connection between room air and outside air, energy recovery vector
① Projection of the supply air setpoint
② Projection of the demand vector
③ Addition of the demand vectors
④ Calculation of the control signal for the energy recovery system

6.8 DEC systems

A skilful combination of adiabatic cooling and adsorptive dehumidification

"DEC" stands for **D**esiccative and **E**vaporative **C**ooling.

The basic thinking underlying DEC technology is to replace the conventional generation of cooling energy in air conditioning systems (using electric compressors) with a method involving air dehumidification functions. To do this, a special method is used, which combines the familiar process of adiabatic cooling with adsorptive dehumidification. Normally, a solid sorbent medium with a proven record is used (e.g. silica gel). The driving force for this process (cf. 5 in Fig. 6-14) is heat, at not too high a temperature, which is often available in the form of waste heat – especially in summer. From the diagram below, it can be seen that this process takes place at a relatively high temperature compared with the air temperature (regenerative heating coils up to 70 °C).

Principle of operation (in summer)

After the normal filtering process, the outside air (e.g. 32 °C and 35 % rh) is dehumidified in an adsorption system (1). This dehumidification is a continuous process, and virtually adiabatic. The heat of adsorption released in this process is emitted into the air flow, thereby heating up the outside air.

The dry warm air is then pre-cooled in a regenerative heat exchanger (2) (which, in winter, is used to preheat the outside air via the extract air). The air, pre-cooled in this way, is then passed through an evaporative humidifier (3) which brings it to the required supply air temperature and humidity.

In a second evaporative humidifier (4) the extract air temperature is reduced to enhance the pre-cooling of the supply air in the heat exchanger (2). The extract air is warmed in this process. The heating coil (5) is then used for reheating, in order to regenerate the adsorption heat exchanger (1). This causes the extract air to cool down, and the humidity to increase. This process is also referred to as "adiabatic desorption."

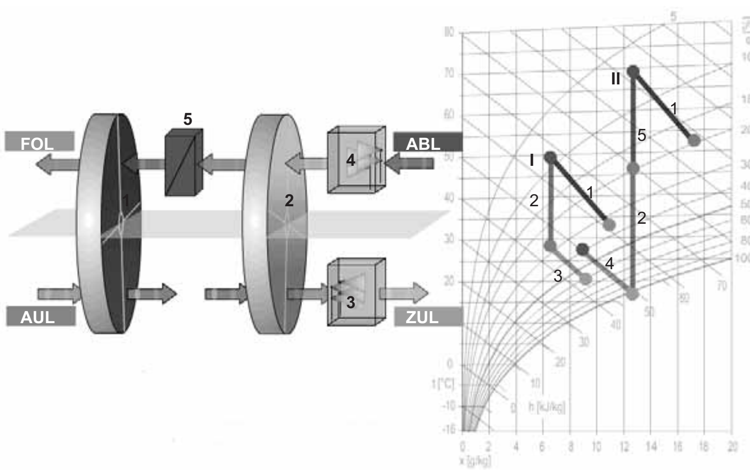


Fig. 6-14 Principle of operation of a DEC system – summer use (Source: Klingenburg)

1	Sorption exchanger (drying of the outside air	
2	Rotary heat exchanger	
3	Supply air humidifier (adiabatic cooling)	
4	Extract air humidifier (adiabatic cooling, e.g. cold vapor generator)	
5	Regenerative heating coil (heats the air to e.g. 70 °C)	
I	State transition (1→2→3);	Outside air (AUL) (supply air (ZUL)
II	State transition (4→2→5→1);	Extract air (ABL) (Exhaust air (FOL)

6.8.1 **Control of DEC systems**

The strategies for the control of DEC systems – matched to the requirements of the specific application – are sometimes highly complex. *Different strategies are often used for heating and cooling*.
The following shows how the system illustrated in Fig. 6-14 is controlled in summer (cooling).
The control system is divided into two control loops:

Control loop 1
Enthalpy at inlet to supply air humidifier

The first control loop controls the enthalpy at the inlet to the supply air humidifer (3) using an evaporative humidifier (4), a rotary heat exchanger (2), a regenerative heating coil (5) and the adsorption system (1).
The first sequence exerts modulating control of the evaporative humidifier (4) between 0...100 %, in which process the rotary heat exchanger (2) operates at maximum speed (100 % energy recovery).
The second sequence exerts modulating control of the regenerative heating coil (5) with the adsorption system (1) operating at its maximum level (100 % dehumidification).
The third sequence (optional) adapts the volume of supply air through step control or speed control.

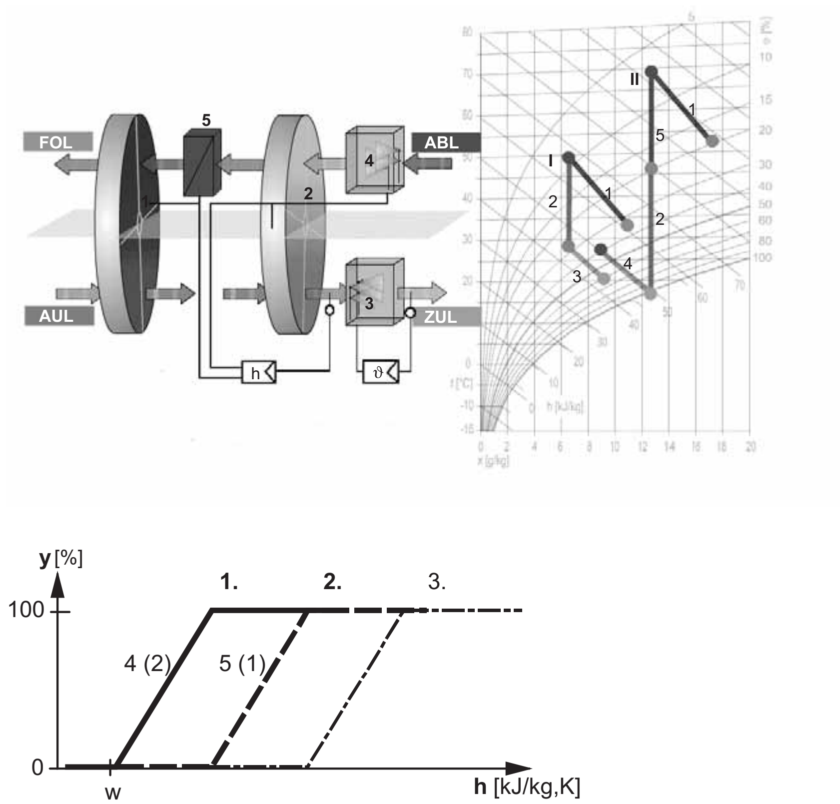


Fig. 6-15 Enthalpy control loop in a DEC system
1st sequence with evaporative humidifier (4) and rotary heat exchanger (2)
2nd sequence with regenerative heating coil (5) and adsorption system (1)
3rd sequence (optional) air volume flow (step control or speed control)

Control loop 2
Supply air temperature

The second control loop provides modulating control of the supply air temperature with the supply air humidifier (3).

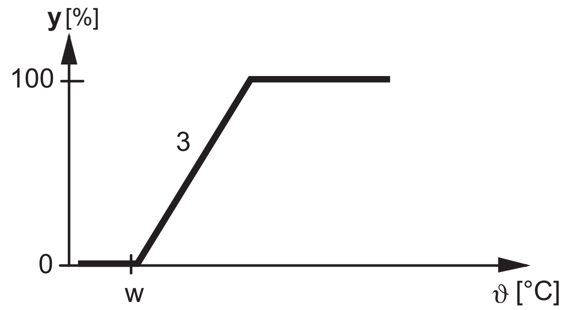


Fig. 6-16 Supply air temperature control loop in the DEC system with the aid of the supply air humidifier (3)

7. Various control functions

7.1 Frost protection

At outside temperatures below 0 °C, there is a risk that the water in the outside air preheater can freeze and damage it. If the water is stationary, ice begins to form immediately. If it is circulating, several degrees minus are required. Therefore, the water temperature in the heating coil must never fall below 0 °C at any location and in any load situation.

The anomaly of water

For information, the volume/temperature relationship of water is shown again here (see also B01HV_en, 2 Physical properties. Of special note is the fact that between 0 °C and 4 °C water first contracts and only then begins to behave normally, i.e. to expand.

	1000 kg water
-1 °C	**Approx. 1090.0 liters**
0 °C	1000.2 liters
2 °C	1000.1 liters
4 °C	1000.0 liters
10 °C	1000.4 liters
20 °C	**1001.8 liters**
30 °C	1004.4 liters
40 °C	1007.9 liters
50 °C	1012.1 liters
60 °C	1017.1 liters
70 °C	**1022.8 liters**
80 °C	1029.0 liters
90 °C	1035.9 liters
100 °C	1043.5 liters

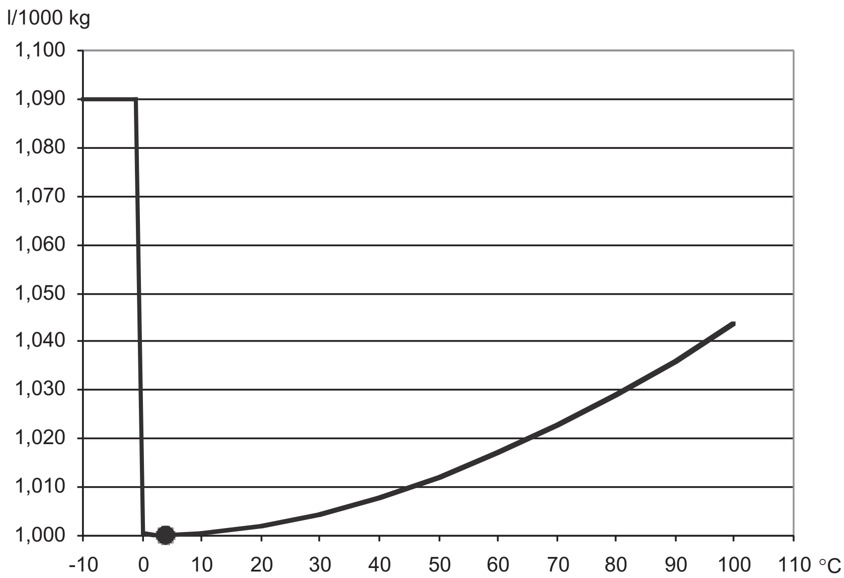


Fig. 7-1 Change in the volume of water as a function of temperature

A frost hazard can be caused by the following:

- Air dampers that do not seal tightly when the system is shut down (causing the heating coil to cool)
- Startup of the system at low outside temperatures when the water in the heating coil has not yet been sufficiently heated
- Disturbances in the heat supply (e.g. burner fault, lack of fuel etc.)
- Disturbances or defects in the system (pump, valve actuator etc.)
- Engineering errors (oversized air heater, incorrect hydraulic circuit – flow control instead of mixing control)
- Low ventilation heat demand in case of large external heat gains in the conditioned space and heat stored in downstream devices (air cooling coil, reheater etc.) causing only slight opening of the control valve

However, the air heater can be protected against frost damage by:

- a frost protection thermostat (in the air or water), or
- two-stage frost protection control (in the air or water), or
- special frost protection startup control.

In the case of prost protection in the water, the circulating pump must be automatically started at outside temperatures below +5 °C.

7.1.1 Frost protection thermostat

Monitoring in the air

The simplest frost protection device consists of a thermostat with a capillary tube sensor. The capillary tube sensor is positioned approximately 5 cm downstream from the heating coil, in loops covering the entire coil surface. The frost protection thermostat switches to "frost hazard" as soon as the air temperature at the capillary tube falls below the selected limit value, e.g. +5 °C, over a length of approximately 30 cm.

Monitoring in the water

In the case of frost protection in the water, a thermostat monitors the water temperature at the heating coil outlet. If the return temperature falls below the set limit, the thermostat switches to "frost hazard".
With both frost protection in the air and in the water, the switchover of the thermostat to "frost hazard" gives rise to the following system actions (Fig. 7-2):

- The heating coil valve is opened to 100 %
- The circulating pump is started (if not already running)
- The supply and extract air fans are switched off
- The outside air and exhaust air dampers as well as fire protection dampers with electronic actuators are closed
- Alarm signaling, locally via a signaling element (e.g. lamp or horn) and to the building automation and control system, if present

If the air temperature rises by the amount of the frost protection thermostat's switching differential, the thermostat enables the system again. In systems with a starting lockout, a local reset button must additionally be actuated manually.
A lockout of this kind should cause the operating personnel to locate the cause of the emergency shutdown and correct any faults before restarting the system.

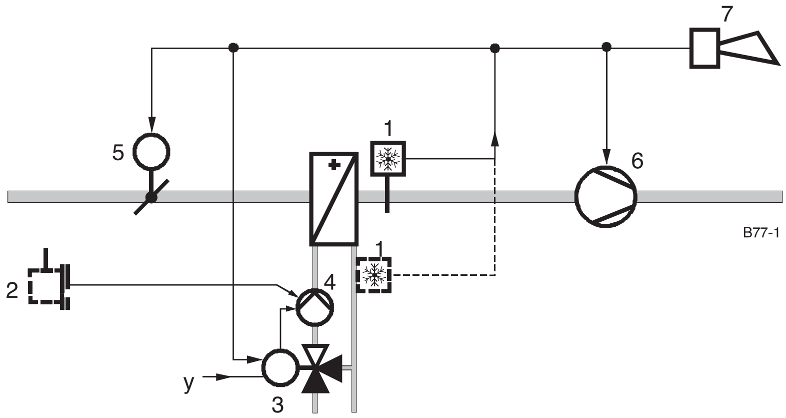


Fig. 7-2 Frost protection thermostat in a simple ventilation system
1 Frost protection thermostat (in the air, alternatively: in the water)
2 Outdoor thermostat per preheater pump
3 Heating coil valve
4 Circulating pump
5 Outside air damper
6 Supply air fan
7 Alarm equipment
y Controller positioning signal

If major secondary damage is expected in case of freezing of the heating coil, security can be increased by combining both frost protection in the air and in the water.

7.1.2 Two-stage frost protection control

Two-stage frost protection control acts on system operation initially with a modulating function (preventive frost protection) and then with a two-position function (frost hazard) as soon as the temperature at the sensing element of the frost protection sensor (1) falls below a set value (Fig. 7-3).

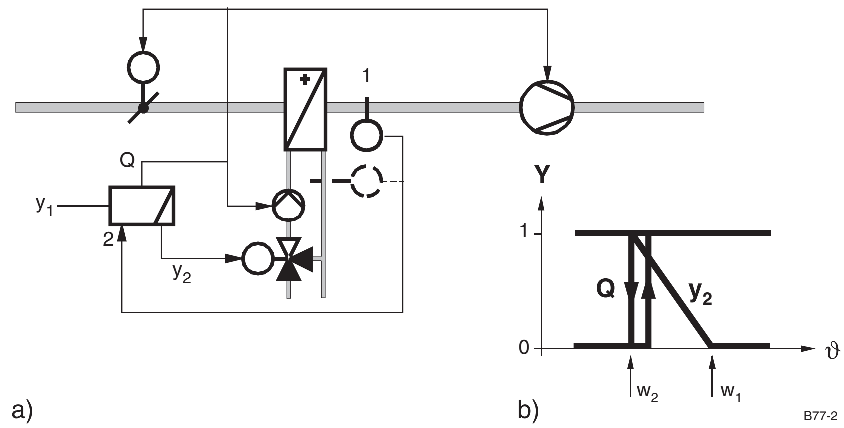


Fig. 7-3 Two-stage frost protection control
a) Schematic
b) Function diagram
1 Frost protection temperature sensor (in air or water)
2 Frost protection controller
y_1 Modulating positioning signal from the frost protection controller
y_2 Modulating positioning signal from the frost protection controller
Q 2-position control signal for fans, dampers and pump
w_1 Frost protection value 1
w_2 Frost protection value 2

Preventive frost protection

The frost protection control acts on the system in the following ways:

Preventive frost protection prevents the unnecessary activation of the "frost hazard" function when the system is shut down (fans off), during startup and during system operation. If the measured temperature falls below the higher frost protection value w_1, the following system elements are continuously commanded by the controller (2):

- Open heating coil valve further (maximum selection between frost protection and temperature control)
- Increase heat recovery output (if present) and mix in more warm air via modulating mixing dampers (if present)
- If a cooling valve is opened, it is closed in order to prevent unnecessary energy consumption

Therefore, preventive frost protection ensures that the frost hazard function is not activated unless the heating coil does not reach the necessary, frost-safe operating temperature in spite of the opening of the control valve or the other modulating heat suppliers.

Frost hazard

If the temperature falls below the lower frost protection value w_2, the following emergency controls are triggered:

- The heating coil valve is opened to 100 % (if not already fully open)
- The circulating pump is started (if not already in operation)
- The supply and extract air fans are switched off
- The outside air and extract air dampers as well as motor-actuated fire protection dampers are closed
- Alarm signaling at the control and monitoring device. The frost hazard can additionally be signaled locally via a signaling element (e.g. lamp or horn) or also externally to a building automation and control system

The frost hazard function is reset when the alarm is acknowledged and the temperature measured by the frost sensor rises above the lower setpoint w_2. However, the valve position remains at 100 % until the temperature reaches the higher setpoint w_1 again.

7.1.3 Special frost protection startup control

If a ventilation or air conditioning system is started at low outside temperatures, there is a particularly high risk that the water in the heating coil will freeze in a very short time because of the sudden inflow of cold air. This problem is more pronounced in the systems with on/off dampers than with modulating dampers and recirculated air mixing.

In a **system with on/off dampers**, frost protection startup control prevents frost damage in the following way (Fig. 7-4): If the system is started (t_1) at an outside temperature of below +8 °C, for example, as acquired by a sensor on the outside wall, initially only the heating coil valve (y_1) is fully opened, and the circulating pump is started. During a time period (Δt_2), which is definable and dependent on the outside temperature, only the preheating of the heating coil occurs, so that there is no longer a frost hazard by the time the fans start. After expiry of the preheating time (Δt_2), i.e. at time (t_3), the air dampers open and the fans start. At the same time, the heating coil valve begins to close according to a definable closing command time (Δt_4) until the heat demand of the temperature controller (y_2) takes over valve positioning.

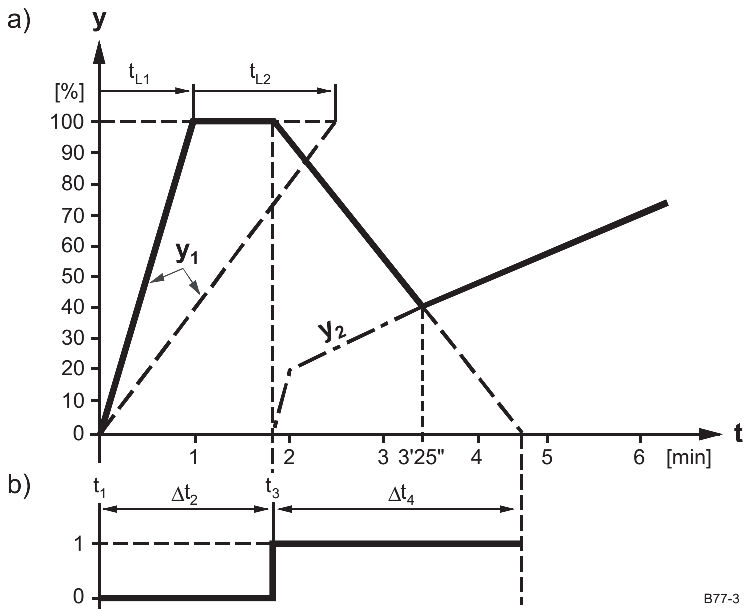


Fig. 7-4 Special frost protection startup control

a)	Preheater valve control
b)	Fan and damper control
t_{L1}, t_{L2}	Actuator running times
t_1	System startup command
Δt_2	Preheating time
t_3	Startup of on/off dampers, fans and closed loop control
Δt_4	Closing command time
y_1	Startup control valve positioning signal
y_2	Temperature controller valve positioning signal

Important When defining a minimum preheating time, the valve actuator running time must be taken into consideration. In normal systems, actuators with opening times of tL_1, = 15...35 s and less frequently t_{L2}, = 30...150 s are used. If the preheating time is below the actuator running time, it is not possible to reach the fully open position.

In a **system with recirculated air mixing**, the fans start immediately when the system is started at an outside temperature below +15 °C, for example. However, the system initially operates in recirculation mode, i.e. the outside air and exhaust air dampers remain closed, and the recirculated air damper is fully open (no minimum outside air proportion). During this period, the hot water circulating pump is running, the heating coil valve is opened, and the cooling valve, if present, is locked out. After a time period which is selectable and partly determined by the outside temperature, the system changer to normal control operation.

7.2 Air filter monitoring

Filters remove solid impurities from outside air and possibly from the extract air, providing for:

- improved indoor air quality and reduced pollution of the conditioned space
- reduced pollution of the system elements and air ducts
- reduced environmental impact

The rising pressure difference across the filter at a constant air flow volume provides an indication of the degree of filter pollution. This can be monitored using a filter monitor or a differential pressure sensor.

7.2.1 Monitoring via filter monitor

As soon as the pressure difference exceeds the limit set at the filter monitor, the filter monitor signals that the filter is dirty. This is indicated as an alarm (Fig. 7-5).

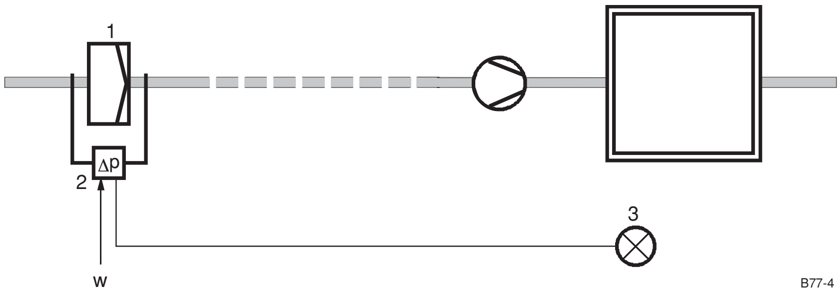


Fig. 7-5 Filter monitoring via filter monitor
1 Air filter
2 Filter monitor (pressostat)
3 Air filter alarm

In variable air flow volume systems, the flow volume control keeps the air flow volume constant in spite of increasing pollution by increasing the fan speed accordingly. If the system is operated with a reduced air flow volume, the fans are run at a disproportionately high speed until the filter monitor is triggered, which consumes energy unnecessarily. In systems of this type and in systems with two or more fan speeds, therefore, optimal filter monitoring via a filter monitor is only possible if the system is operated at the maximum flow volume at least once a day.

7.2.2 Monitoring via differential pressure sensor

In variable air flow volume systems, the pressure drop across the filter does not only depend on the degree of filter pollution but also on the air flow volume. Additionally, since the pressure drop changes exponentially with respect to the flow volume, a reduction in flow volume from 100 % to 50 % produces a decrease in the pressure drop from 100 % to 25 % (see also B01HV_en, 6 Overview air conditioning systems).

In order to provide effective filter monitoring in such systems even at reduced flow volumes, it is recommended that the degree of air filter pollution should be monitored via a differential pressure sensor with compensation of the limit setpoint for filter pollution based on the current air flow volume. This compensation is accomplished using a shift controller (5), taking the air velocity (4) as the reference variable (Fig. 7-6).

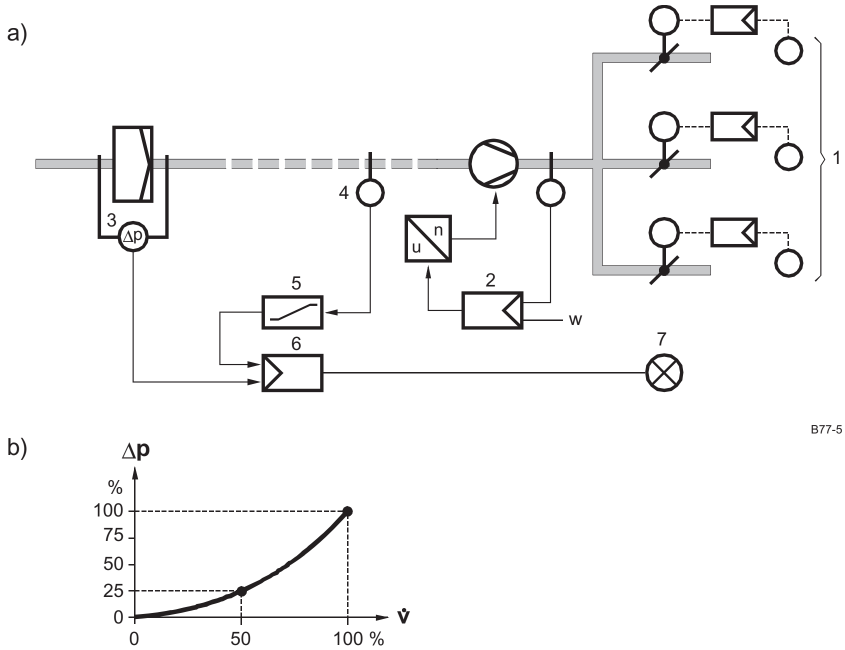


Fig. 7-6 Filter monitoring via differential pressure sensor in case of variable air flow volume

a) Filter monitoring schematic
b) Shift controller function diagram
1 Zones with variable supply air flow volume
2 Supply air pressure control with fan speed control
3 Differential pressure sensor for filter monitoring
4 Air velocity sensor
5 Setpoint shift controller
6 Differential pressure controller for filter monitoring
7 Air filter alarm

7.3 Ventilation and air conditioning systems with electric heating coils

In principle, these systems are controlled in the same way as systems with LTHW heating coils. In this case, however, output is controlled with a multi-step controller, digital step controller or current valve, i.e. the output signal from the temperature controller is converted into a control signal suitable for the electric heating coil.

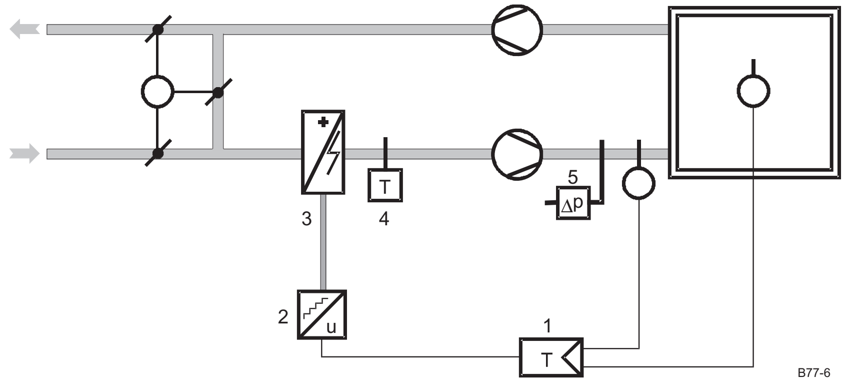


Fig. 7-7 Electric heating coil with power control via step controller
1 Temperature controller
2 Step controller
3 Electric heating coil (multistep)
4 Safety limit thermostat
5 Air flow switch

A multi-stage electric heating coil, i.e. one in which the heater elements are wired as separate power units (e.g. in sevenths:1/7, 2/7 and 4/7) can be connected directly to a multiple or digital step controller. In this case different power units are switched (sometimes in combination) depending on the situation in the plant.

Electric heating coils can also be operated with a current valve. The current valve regulates the required power with a contact-free power switch (triac). The current valve itself is driven with a pulse-pause output signal. Current valves are capable of direct switching of only a certain level of power. For higher power levels, several current valves must be used (refer to the manufacturer's literature).

In the design of electric heating coils for comfort systems, it should be ensured that the temperature difference per power step does not exceed about 3 K. This allows uncomfortable drafts within the conditioned space to be avoided.

Since the heating elements of the electric heating coil can reach very high temperatures if the air flow fails, special attention should be paid to the fire hazard in all applications. Therefore, air flow monitoring (air flow switch, air-vane switch or barometric cell), a safety limit and a fan overrun time should be implemented as a safety circuit as follows:

- The air flow switch prevents the electric heating coil from being switched on before air flow is signalized, or it switches it off if there is no longer an air flow due to a defect or the shutdown of the fan. A definable switch-off delay prevents a shutdown with fault from being triggered by brief air flow fluctuations

- The safety limit thermostat monitors the temperature of the electric heating coil, and switches it off if overheating is detected.
- Fan run-on: In systems with electric heating coils, the fans should remain in operation for a definable duration after the shutdown of the system in order to prevent an excessive buildup of heat in the vicinity of the electric heating coil.

In more complex systems with two-speed supply air fans, step monitoring ensures that the heat output of the electric heating coil is reduced to 50 % for a definable duration on switchover from the higher to the lower speed. This prevents the excessive temperature rise that would otherwise occur in the supply air when the air flow volume is reduced. After expiry of the defined duration, the temperature controller resumes the output control of the electric heating coil.

7.4 Displacement ventilation

In rooms with primarily cooling load, the continually increasing demands on ventilation systems regarding freedom from drafts and the removal of heat and pollutants can be largely met with displacement ventilation. However, any heating load must be covered by static heating surfaces.

In displacement ventilation, the conditioned air is injected at floor level in a laminar (low turbulence) fashion and at slightly lower than room temperature (Fig. 7-8). The supply air temperature should, on the one hand, be a maximum of 2...3 K lower than the room temperature in offices and up to 8 K lower in factories, and on the other, it should not be less than 21 °C in offices or 17 °C in industrial spaces (uncomfortably cold feet). The outlet speed should be approximately 0.2 m/s in offices and up to 0.6 m/s in other applications. This causes a "pool of fresh air" to form in the occupied zone. The thermal updrafts on people and devices cause the air to rise to the ceiling area, where it is extracted again. Because the outside air only rises in the vicinity of heat sources, the heat and material load is dissipated exactly where it occurs and is not distributed throughout the room. This allows a high air quality to be achieved with relatively small air quantities. The normal air change coefficient is between 1...4 h^{-1}.

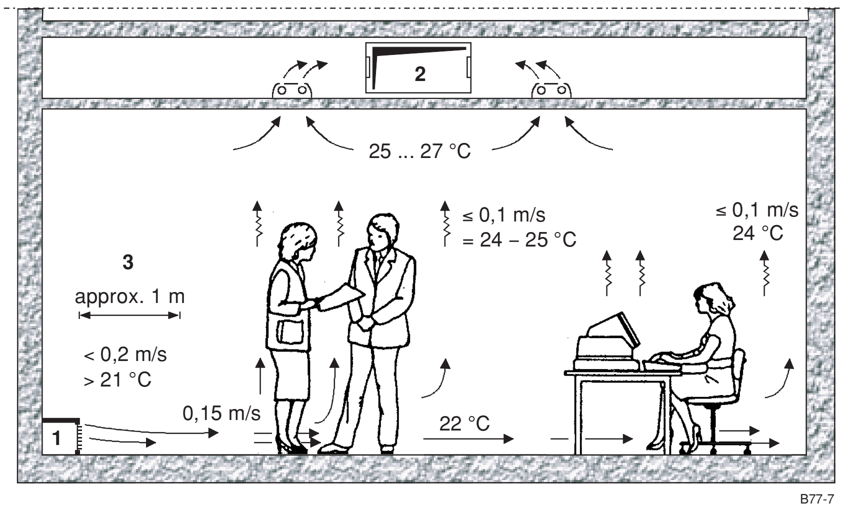


Fig. 7-8 Principle, temperatures and air velocities of displacement ventilation
1 Displacement air outlet
2 Extract air duct
3 Proximity zone

Where displacement ventilation alone is used in comfort applications, the supply air temperature is generally limited to approximately 2...3 K below the room temperature. Therefore, only relatively small cooling loads can be handled, or correspondingly large air quantities are required for larger cooling loads. Additionally, thermal comfort cannot be guaranteed in the direct vicinity of the supply air outlet, so a certain distance from occupied zones is required, i.e. the outlets should not be installed in the vicinity of occupied zones.

Displacement ventilation systems are especially suitable for rooms that do not contain vastly different loads or where indoor air quality plays a decisive role (factory buildings, sports halls, hotels, theatres, schools, restaurants). They satisfy high demands on comfort, especially in combination with chilled ceilings.

Control

In a displacement ventilation system, the supply air temperature (injection temperature) is normally controlled (see also Chapter 1). If required, the supply air temperature control setpoint can also be room temperature (or outside air temperature) compensated.

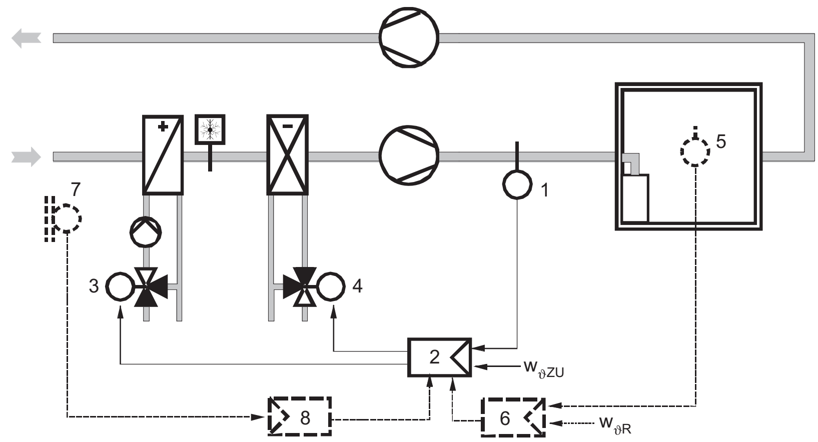


Fig. 7-9 Supply air temperature control in a displacement ventilation system
1 Supply air temperature sensor
2 Supply air temperature controller
3 Heating valve and actuator
4 Cooling valve and actuator
5 Room temperature sensor
6 Room temperature setpoint shift controller
7 Outdoor temperature sensor
8 Outside temperature setpoint shift controller

The air velocity shown in Fig. 7-8 at the outlet (e.g. 0.15 m/s) is not controlled: instead it must be achieved in the planning and implementation stages, by ensuring that the outlets are correctly sized.

7.4.1 Combined displacement ventilation and chilled ceiling

A combination of displacement ventilation and chilled ceilings is used in situations where large cooling loads need to be dissipated. The displacement ventilation is used to renew the indoor air, and the chilled ceilings dissipate the cooling load. This makes it possible to satisfy very high expectations of comfort.

The displacement ventilation system is controlled at a constant supply air temperature. A minimal cooling output can also be introduced into the room at times (e.g. in winter, in the event of internal heat gains). The chilled ceilings are controlled via the room temperature control in sequence with the room heating (where this is part of the control system).

Dew point monitoring

If chilled ceilings are used, it is important to ensure that the temperature of the chilled ceilings and pipes – and hence, the temperature of the cooling water – is always above the dew point temperature of the room. Otherwise, condensation will occur.

There are various methods of dew point monitoring:
- Control of the cooling-water flow temperature based on the outside temperature
- Local dew point monitoring in the room (on/off)
- Central control of the cooling-water flow temperature based on a critical room
- Local control of the cooling-water flow temperature based on conditions in the room

7.4.1.1 Control of the cooling-water flow temperature based on the outside temperature

The cooling-water flow temperature is adjusted throughout the building on the basis of the dew point of the outside air. This is a cost-effective and efficient method of dew point monitoring. However, it does involve working with a relatively large safety margin, which can lead to a reduction in the useful cooling output in individual rooms.

A further adjustment is also possible based on a calculation of the dew point in a reference room or in the extract air. The humidity loads can be registered centrally or for individual zones, provided that the humidity load is distributed evenly. The outcome is a reliable flow temperature for the cooling water.

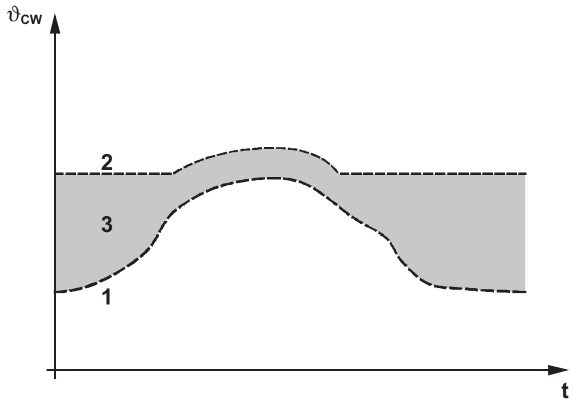


Fig. 7-10 Central control of the cooling-water flow temperature on the basis of the outside temperature
1 Dew point temperature of the outside air
2 Cooling-water flow temperature
3 Safety margin

7.4.1.2 Local dew point monitoring in the room (on/off)

Monitoring is carried out locally with a condensation detector in each room. The chilled ceiling is switched off if the dew point is exceeded. This is a cost-effective way of providing individual room-by-room protection from condensation. The disadvantage of this solution is that if the dew point is exceeded, cooling in the room is disabled.

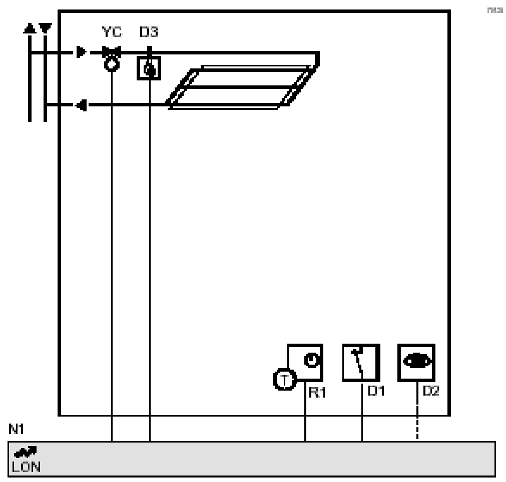


Fig. 7-11 Chilled ceiling with dew point monitoring (example: Siemens DESIGO RX)

D3 Dew point sensor	R1 Room unit with temperature sensor
YC Cooling valve (modulating)	D1 External contact (e.g. window)
N1 Individual room controller	D2 Occupancy sensor

The condensate detector must be installed at the coldest point (cold-water inlet) and must be well able to register the humidity of the room air. This solution is not recommended in the cases of ceilings with sensitive surfaces.

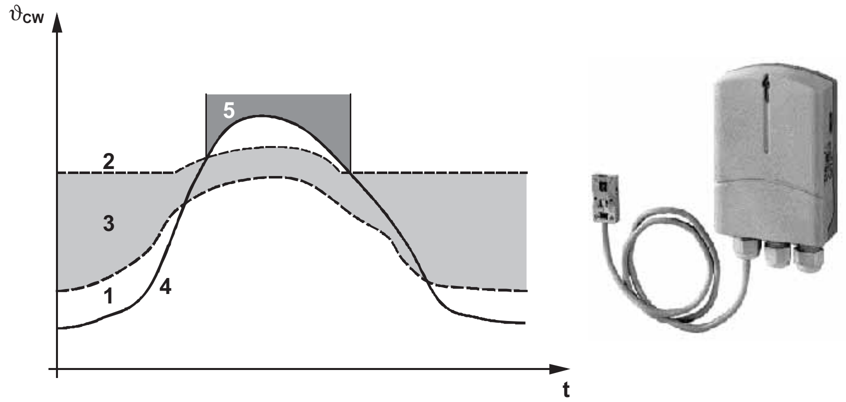


Fig. 7-12 Local dew point monitoring with condensation detector (e.g. Siemens condensation monitor QFX 21, right)
1 Dew point temperature of the outside air, extract or of a reference room
2 Centrally controlled cooling-water flow temperature
3 Safety margin
4 Dew point temperature in a critical room
5 Cooling disabled in critical room by condensate detector

7.4.1.3 Central control of the cooling-water flow temperature based on a critical room

The dew point is monitored in sequence with the central control of the cooling-water flow temperature and the local dew point calculation. The permissible cooling-water flow temperature is calculated from the room humidity and temperature individually for the critical room and – if required – raised centrally. For the critical room, the cooling-water connection must be implemented with a variable-temperature hydraulic circuit (e.g. mixing or injection circuit).

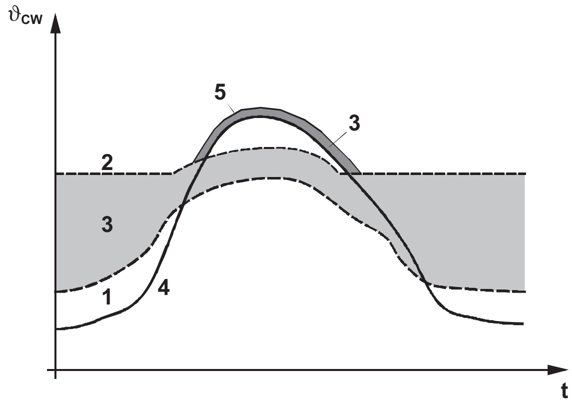


Fig. 7-13 Central control of the cooling water flow temperature based on the room air temperature
1 Dew point temperature of the outside air, extract air or of a reference room
2 Centrally controlled cooling-water flow temperature
3 Safety margin
4 Dew point temperature of a critical room
5 Cooling-water flow temperature calculated for the critical room

7.4.1.4 Local control of the cooling-water flow temperature based on room conditions

The dew point is monitored individually in each room. The permissible cooling-water flow temperature is calculated in each room on the basis of the temperature and room humidity, and – if necessary – raised in this room. The cooling-water connection is implemented with a variable-temperature hydraulic circuit (e.g. mixing or injection circuit).

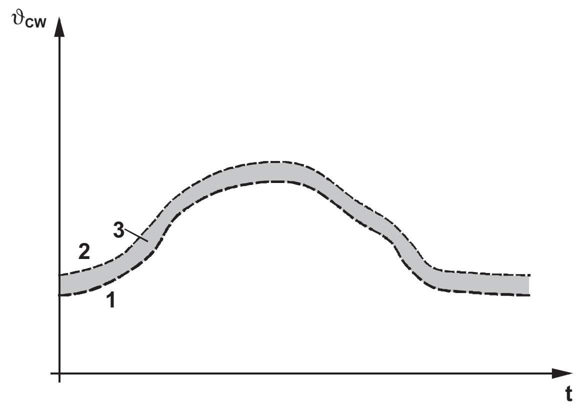


Fig. 7-14 Local control of the cooling-water flow temperature on the basis of conditions in the room
1 Dew point temperature in the room
2 Cooling-water flow temperature
3 Safety margin

7.5 Pressure and flow volume control in ventilation systems

In many large-scale systems (e.g. central air treatment with individual room control) a frequent, very important requirement is that variable air flow volumes can be transported at constant pressure. Otherwise, the differing duct pressures arising from throttling or cutting off the air flow in individual rooms or zones would cause a corresponding increase in the supply air flow volume in other rooms or zones, with the associated draft and noise nuisance. In other systems, e.g. for certain hospital rooms, production facilities, office buildings etc., a higher or lower pressure than that of the outside air or neighboring rooms is required.

The demand-driven control of an air pressure difference or an air flow volume can, for example, be accomplished via the following manipulated variable changes:

- Stepless fan motor speed control
- Pole-changing motor (stepwise speed change)
- Inlet guide vane control: a greater or lesser swirl, and therefore a greater or lesser pressure drop, is generated in the air flow by variable guide vanes in front of the fan rotor inlet
- Rotor blade control (for axial fans only): the angle of attack of the rotor blades is continuously variable during running via a gearbox that is incorporated in the rotor hub (costly solution for large industrial fans)
- Parallel operation of several smaller fans instead of a single large fan. One of the fans can be additionally equipped with stepless throttle control, whilst the others are taken on and offline according to load
- Adjustable bypass damper (short circuit) across the fan: Depending on the air damper position, a variable quantity of the air delivered by the fan is returned from the fan outlet (pressure side) to the fan inlet (suction side)
- Variable extract air dampers: The air resistance of the extract air duct varies according to the position of the extract air dampers. This results in variable pressurization of the ventilated space

7.5.1 Supply air pressure control

If it must be possible to throttle or shut off individual sections of a branching duct network without affecting other operational systems, the supply air duct must be maintained at a specific pressure or at a specific pressure differential with respect to the environment (Fig. 7-15).
The modulating controller (1) compares the pressure difference (duct pressure with respect to the environment) acquired by the sensor (2) against the setpoint. In case of a deviation, the controller corrects the delivery pressure of the fan (3) via the rotor speed, the guide vanes or (alternatively) the bypass dampers (4).

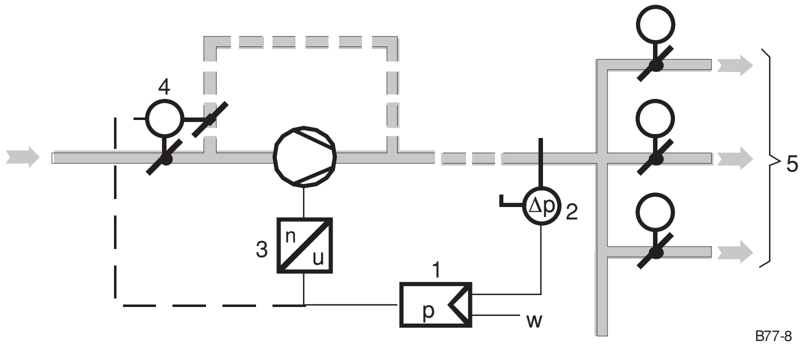


Fig. 7-15 Supply air pressure control in a branching duct network
1 Supply air pressure controller
2 Differential pressure sensor
3 Fan speed control
4 Bypass damper with modulating actuator (alternative)
5 Different zones

There are now commercially available variable speed drives (3) which incorporate the pressure controller (1) as a built-in component, and which allow direct connection of the differential pressure sensor (2).

Fig. 7-16 Variable speed drives with built-in pressure control (e.g. Siemens SED2)

7.5.2 Positive or negative room pressure control

With this type of control, a positive or negative pressure is generated in the ventilated space with respect to the outside air and neighboring rooms. A positive or negative pressure can be generated in an enclosed space, for example via different delivery rates of the supply air and extract air fans, varying the duct air flow resistance via dampers etc.

If a room is kept at a **positive pressure**, the ingress of undesirable polluted air through leaks is prevented (Fig. 7-17). Positive pressure control is used, for example, in laboratories and production rooms for high-precision electronic, optical and mechanical devices as well as in hospital operating theatres (to prevent infections). A certain positive pressure is created by keeping the supply air flow constant while the controller (1) throttles the extract air flow via the extract air damper (3) or varies the guide vanes of the fan (4) according to the desired positive room pressure (sensor (2)).

Negative pressure control of an enclosed space prevents bad air from spreading into adjacent rooms. Therefore, it is primarily used in rooms with heavy air pollution from gases, vapors or smells, e.g. in kitchens, toilet facilities, cloakrooms, laboratories, production rooms, battery rooms etc. In hospital operating theatres, the spread of bacteria is prevented by negative pressure operation. In order to create a negative pressure, the extract air flow is kept constant, while the supply air flow is throttled according to the desired negative pressure in the ventilated space.

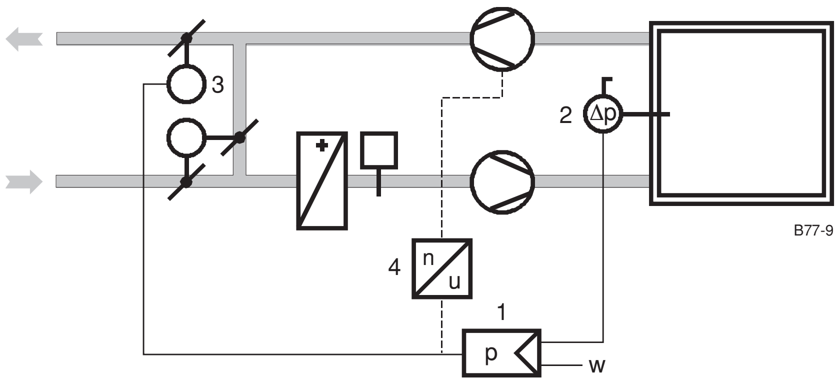


Fig. 7-17 Positive pressure control in an enclosed space
1 Pressure controller
2 Differential pressure sensor
3 Extract air damper with modulating actuator
4 Power controlled extract air fan (alternative)

Aseptic / septic

In the context of positive and negative pressure control, reference is often made, e.g. in hospitals, to "aseptic" and "septic" rooms.

"Aseptic" means sterile, or germ-free. Aseptic rooms are rooms that are free of germs, and into which the infiltration of contaminants or pollutants must be prevented. ⇨ This is where positive pressure is required.

"Septic" comes from the Greek word sepsis and means putrid or decayed. "Septic rooms" are rooms which are polluted by pathogens, bacteria etc., which must not be allowed to seep out of the room. ⇨ Negative pressure is required here.

7.5.3 Air volume control

Air flow volume control is used, for example, in systems where the air flow volume should remain constant in spite of an increasing degree of filter contamination (= pressure drop) or where one air flow volume fluctuates and another must be varied in proportion to it. An example could be a production facility in which the extract air flow volume frequently changes because of the use of numerous extraction devices, and the supply air flow volume must be varied in proportion to it in order to maintain a slight positive or negative pressure although the building is not airtight (Fig. 7-18).

Cascade control provides a stable control action in this case as well. The measured variable for the air flow volume in the supply air and extract air duct is the dynamic pressure, i.e. the difference between total pressure and static pressure. Differential pressure sensors with diaphragms to register the pressure differential serve as sensing devices. In the cascade control (Fig. 7-18), differential pressure sensor (2) acquires the supply air flow volume and differential pressure sensor (3) the extract air flow volume. The supply air flow volume controller (1, slave controller) controls the supply air flow volume according to the extract air flow volume, i.e. if the extract air flow volume changes, the extract air flow volume controller (4, master controller) changes the set-point of the supply air flow volume controller according to the defined cascade authority (see also chapter 1, section 1.5). The final control element of the control loop is, for example, a fan speed control device (5) or a damper actuator. Since the extract air flow volume serves as the reference variable for the supply air flow volume, pressure influences from the outside or from adjacent rooms have no effect.

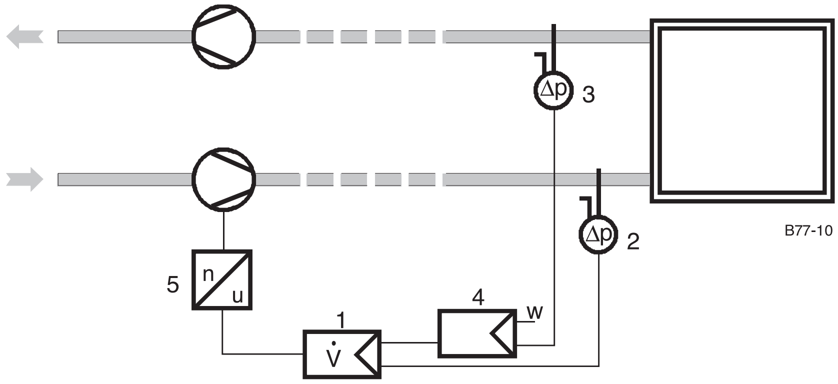


Fig. 7-18 Supply air flow volume control based on extract air flow volume
1 Supply air flow volume controller
2 Supply air differential pressure sensor
3 Extract air differential pressure sensor
4 Extract air flow volume controller
5 Supply air fan speed control

7.6 Night purge cooling

During periods with high daytime temperatures, night purging is used to reduce cooling energy consumption, especially in buildings with large heat storage capacities, by precooling the rooms with cool night air for the following day. The night purging function must be adapted to the building in order to guarantee its energy-optimized operation. This adaptation is made on the controller (Fig. 7-19) by appropriate adjustment of the times and temperature limit values for the activation and deactivation conditions.

Switch-on conditions

Night purging is activated if the following conditions are fulfilled simultaneously:
- Night purging is enabled by the time switch program
- The outside air temperature is above the defined low limit of 12 to 14 °C
- The room temperature is above the set high limit
- The difference between the room temperature and the outside air temperature is greater than the defined difference

Switch-off conditions

Night purging is deactivated:
- By the time switch program, or
- At the earliest, after expiry of the defined minimum operating time, if one of the activation conditions listed above is no longer met

In the night purging operation mode:
- On/off air dampers are fully opened
- Modulating dampers are driven to the maximum position (recirculated air dampers closed)
- Single speed fans are switched on
- Two-speed fans are operated at the higher speed

Not activated are heating coil, heat recovery and closed-loop control

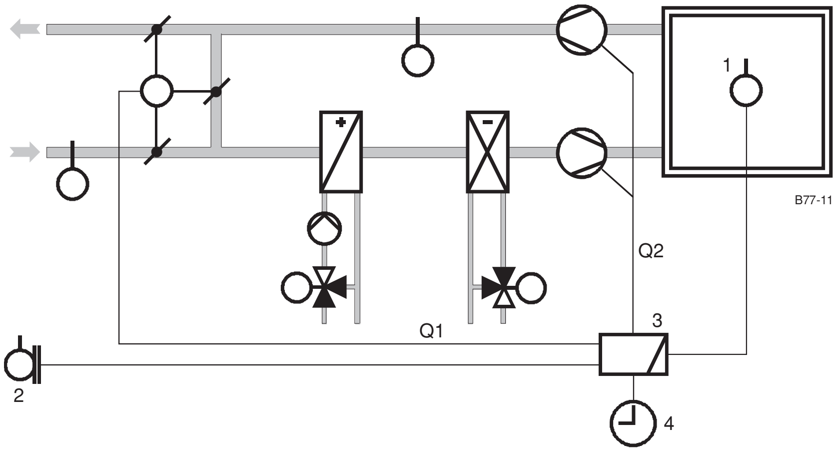


Fig. 7-19 Principle of night purging
1 Room temperature sensor
2 Outdoor sensor
3 Controller or control program
4 Time switch
Q1 Air damper control signal
Q2 Fan control signal

7.7 Fire control procedure

If the fire alarm system signalizes a fire in the building, the fire control procedure assumes control of system operation. This includes the following functions:

- Deactivation of the system and system elements
- Signalizing the fire at the process unit, to the fire alarm control and indicating equipment and externally via an alarm horn or lamp
- Smoke extraction from the building after the fire

7.7.1 Fire protection cutoff

The fire alarm switches the system to fire protection cutoff mode. This means that, depending on system configuration, the following switching operations are performed:

- Deactivation of the fans
- Closing the outside air and exhaust air dampers
- Closing the motor driven fire protection dampers

7.7.2 Smoke extraction mode

The smoke extraction function provides for the removal of smoke and heat from inside the building after a fire. Only when it is in fire protection cutoff mode can the system be switched to smoke extraction mode via the smoke extraction command of the fire service switch. Depending on the system, the following smoke extraction possibilities are available:

In systems without overpressure or underpressure hazard:

- Smoke extraction with the supply air fan only
- Smoke extraction with the extract air fan only
- Smoke extraction with the supply air and extract air fan simultaneously

In systems with overpressure or underpressure hazard:

- Smoke extraction with the supply air and extract air fan simultaneously

Depending on the smoke extraction concept, modulating dampers, appropriate on/off dampers and fire-protection dampers are opened, and the appropriate fans are activated.
Smoke extraction operation with speed-controlled fans in VAV systems is only possible if the throttling dampers of the air flow volume controls in the individual zones are fully opened. In such systems, therefore, a switching command to open the throttling dampers must be triggered in smoke extraction mode.

8. Air retreatment control

8.1 General

The purpose of air retreatment is to adjust the supply air to the comfort needs of individual rooms or room zones in a building. The overall goal is to optimize energy consumption. This goal is primarily achieved by providing only as much comfort as necessary, when and where it is used or requested. For example, there is no point maintaining comfort levels in office or hotel rooms when they are unoccupied.

The different heat gains in individual rooms or zones require individual adjustment of heating or cooling output by varying either the temperature or flow volume of the supply air. How these adjustments are made by control means in common system types is explained in this chapter. The functional principle of these systems is explained in the training module B01 "Introduction to HVAC technology," chapter 7 "Introduction to ventilation and air conditioning."

8.2 Single-duct system with zone retreatment

The centrally pretreated air is retreated for each room or zone according to the desired room air state. The retreatment can include the reheating, recooling, redehumidification or rehumidification functions.

In single-duct systems with terminal zone retreatment (Fig. 8-1), the retreatment takes place close to the respective zone. The positioning signals of the room or zone controllers (7) act directly on the control elements of the retreatment components (3+4) according to individual demand.

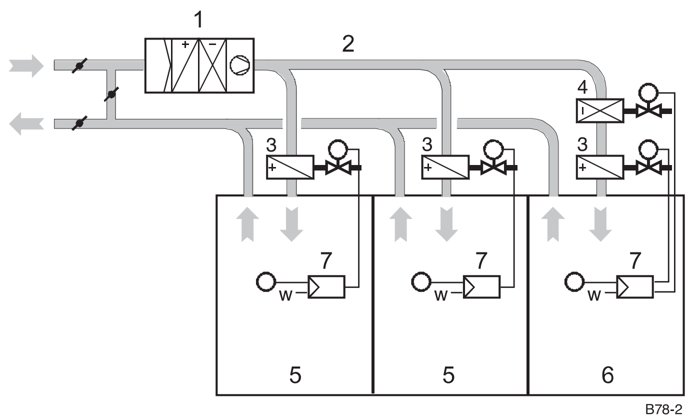


Fig. 8-1 Single-duct system with terminal zone retreatment

1	Central air treatment plant
2	Supply air duct
3	Air heating coil as a retreatment
4	Air cooling coil as a retreatment
5+6	Room or room zone
7	Room or zone temperature controller

In single-duct systems with central zone retreatment (Fig. 8-2), the retreatment takes place immediately downstream from central air treatment. The positioning signals of the room or zone controllers act directly on the control elements of the retreatment components according to individual demand. However, the positioning signals must be wired from the individual rooms or zones to the control panel of the central air treatment plant. In order to avoid energy wastage, the supply air temperatures required simultaneously by the individual zones should not diverge greatly.

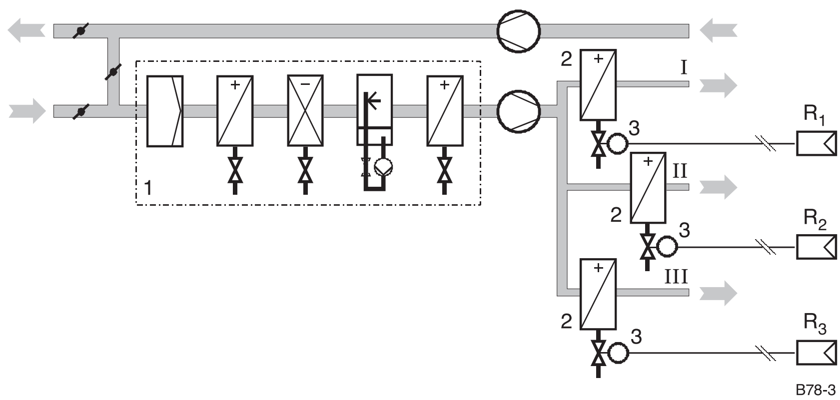


Fig. 8-2 Single-duct system with central zone retreatment

1	Central air treatment plant
2	Zone reheating coil
3	Zone heating valve
I...III	Zone supply air
R1...R3	Zone temperature controllers

8.3 Multizone system with multizone central air treatment plant

First, the outside/recirculated air mixing, filtration and preheating of the entire air quantity takes place in the central air treatment plant (Fig. 8-3). Downstream from the supply air fan, the supply air flow is divided into two partial flows (Fig. 8-4). One partial flow is directed through the reheater, the other through the cooler. The individual supply air temperature required for each zone is then provided by mixing the cool and warm air flows in the subsequent zone dampers (5). The zone dampers are arranged vertically with a cool air and a warm air damper on common damper shaft. The cool air damper is at a 90° angle to the warm air damper, so that the warm air damper is fully open when the cool air damper is closed.

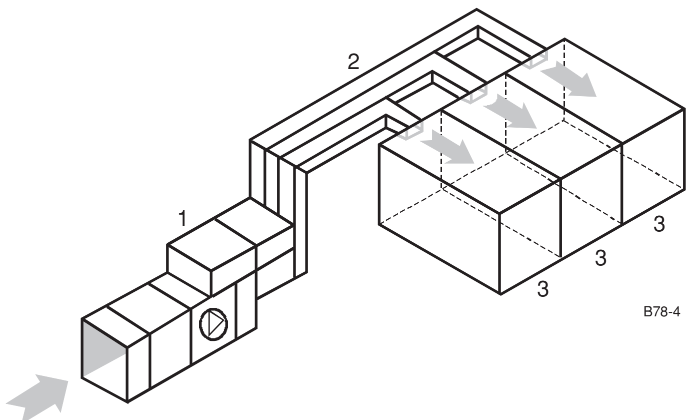


Fig. 8-3 Multizone system

1 Multizone central air treatment plant

2 Supply air ducts

3 Different zones

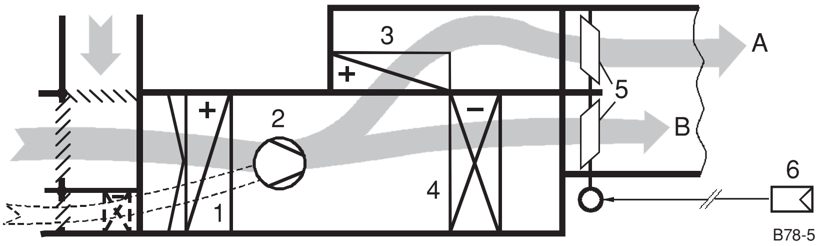


Fig. 8-7 Functioning of a multizone system

1	Preheater	5	Zone dampers
2	Supply air fan	6	Zone temperature controller
3	Reheater	A	Warm air
4	Air cooling coil	B	Cool air

With regard to heating and cooling energy consumption (mixing losses), it is advantageous if the supply air temperatures for the individual zones differ only very slightly (< 5 K).

8.4 Dual-duct systems

Warm and cool air is mixed according to the requirements of the individual rooms in specially designed mixing boxes that are installed in the rooms (behind a false ceiling). The mixing ratio is controlled by the individual room temperature controllers.

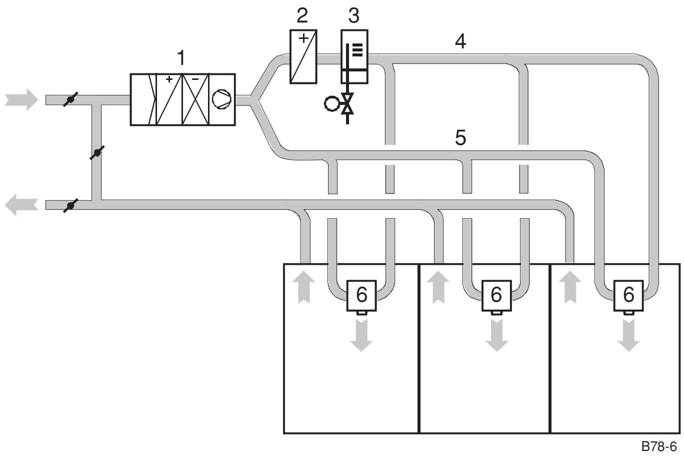


Fig. 8-5 Dual-duct system with dehumidification of the entire supply air flow
1 Central air treatment plant
2 Reheater
3 Steam humidifier
4 Warm air duct
5 Cool air duct
6 Mixing boxes

The supply air temperature setpoints do not remain constant; the warm air temperature corresponds to the highest supply air temperature setpoint and the cool air to the lowest supply air temperature setpoint of all connected room temperature controllers at any given time (Fig. 8-6). Modern digital technology makes it possible to interrogate the current values via a building bus and select the maximum and minimum values in each case.

This allows mixing losses to be reduced. Rooms with maximum cooling load receive only cool air, those with maximum heating load only warm air, and those with partial load a mixture of cool and warm air.

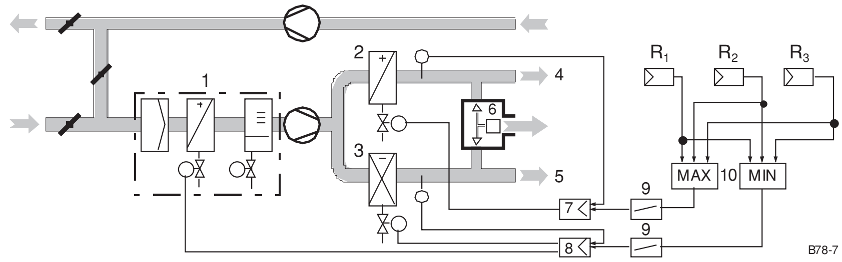


Fig. 8-6 Dual duct system with partial dehumidification of the supply air

1	Central air treatment	6	Mixing boxes
2	Reheater	7	Warm air controller
3	Cooling coil	8	Cool air controller
4	Warm air duct	9	Setpoint shift controller
5	Cool air duct	10	Max./min. selection of the positioning signals
R1...R3	Room or zone temperature controllers		

The air for the cool air duct is conditioned to the required temperature and dehumidified in the central air treatment plant, that for the warm air duct is heated and possibly humidified. The air cooling coil configuration shown in Fig. 8-5 provides for controlled dehumidification of the total supply air flow. In combination with the steam humidification in the warm air duct, this configuration gives rise to a full air conditioning system with room temperature and humidity control. However, this comfort comes at the cost of relatively high energy consumption for dehumidification and subsequent reheating of the supply air, so it is now only approved in special cases. Therefore, the configuration shown in Fig. 8-6 with only partial, uncontrolled dehumidification of the cool air flow via condensation is the standard solution for normal comfort requirements.

Fig. 8-7 shows the principle design of a dual-duct mixing box. The mixing boxes are equipped with an air mixing facility (valves or dampers). Additionally, they have a mechanical flow volume regulator which keeps the supply air flow volume constant even in case of pressure fluctuations in the supply air ducts. Mixing boxes with variable cool air flow volume are also available (Fig. 8-8).

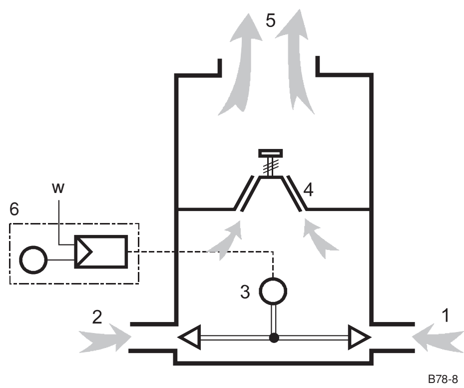


Fig. 8-7 Design of a dual-duct mixing box

1 Cool air inlet
2 Warm air inlet
3 Air mixing valve with actuator
4 Constant flow volume regulator
5 Supply air
6 Room temperature controller

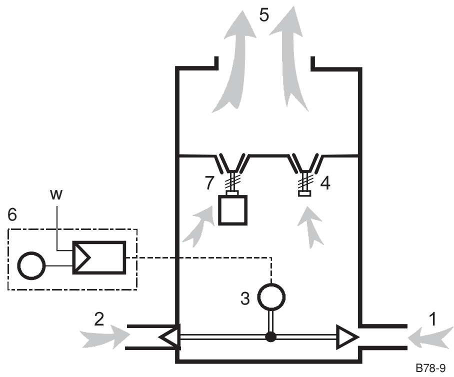


Fig. 8-8 Dual-duct mixing box with variable cool air volume control
1 Cool air inlet
2 Warm air inlet
3 Motorized air mixing valve
4 Constant flow volume regulator
5 Supply air
6 Room temperature controller
7 Cool air flow volume controller (50 to 100 %)

Room temperature control is accomplished as shown in Fig. 8-9. The PI-controller (1) compares the room temperature acquired by the sensor (2) against the setpoint. In case of deviation, it varies the cool/warm air mixing ratio via the actuator (3), and therefore also the supply air temperature, until the control deviation is corrected.

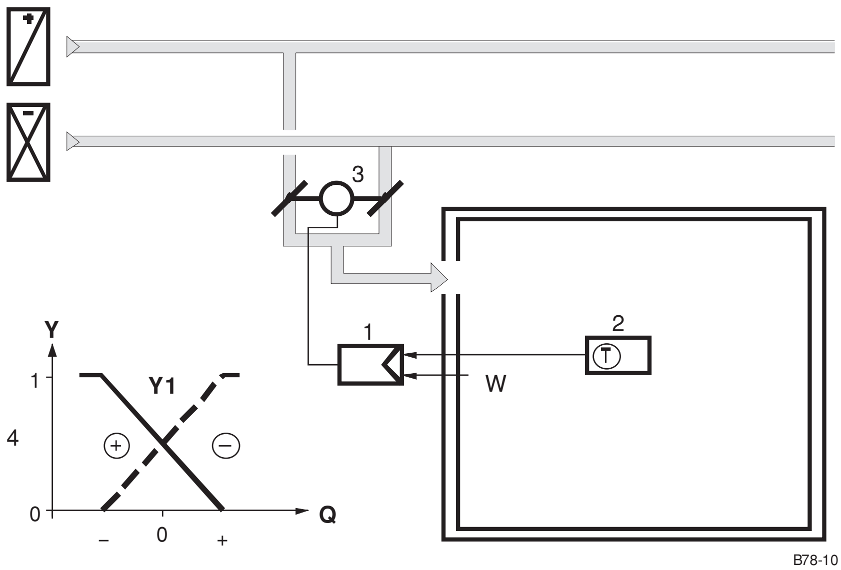


Fig. 8-9 Individual room temperature control with a dual-duct system
1 Room temperature controller
2 Room unit with temperature sensor
3 Actuator for mixing dampers or valves
4 Room temperature control function diagram

If communicating, digital room temperature controllers are used, the building automation and control system provides the following additional functions:

- Predefined setpoints for comfort or economy operation
- Predefinition of all control parameters
- Interrogation of the current positioning signals for shifting the duct temperature setpoints
- Monitoring of all room temperature control loops in variable air flow volume (VAV) cooling systems

8.5 Operating principle of VAV systems

A VAV (Variable Air Volume) system is basically a cooling system, so it must be combined with an appropriate heating system (radiator or floor heating) for heating operation. All cooling is provided by the supply air. In order to do so, the supply air temperature is maintained at a predefined setpoint, and the room temperature is kept at the desired setpoint by varying the supply air flow volume. Zoning of the building can be dispensed with, because the supply air flow volume can be individually adjusted to the sensible cooling load in each room. Additionally, if suitable air outlets are used, the temperature difference between room and supply air can be considerably greater than in conventional systems.

In the VAV system shown in Fig. 8-10, the centrally treated supply air is transported via a single-duct system to the conditioned rooms, where it is injected at a variable flow volume depending on the individual cooling load.

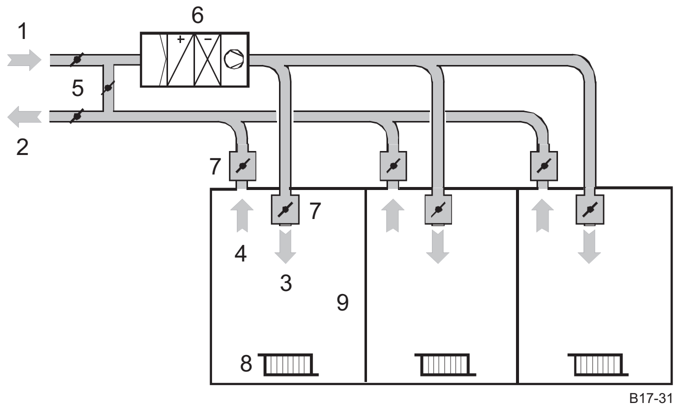


Fig. 8-10 Air cooling system with variable air volume (VAV)

1 Outside air
2 Exhaust air
3 Supply air
4 Extract air
5 Air dampers
6 Central air treatment plant
7 VAV Box (supply and extract)
8 Basic heating system
9 Room

8.5.1 Individual room temperature control with VAV

Stable room temperature can only be achieved with P+PI cascade control. If the room temperature control acted directly on the flow volume actuator, it would only work in on/off mode because of the inertia of the room control loop, giving rise to major room temperature fluctuations and unacceptable drafts.

In the case of cascade control as shown in Fig. 8-11, the room temperature controller (1) controls the setpoint of the air velocity controller (3) which can very quickly detect the effect of the actuating action (5) via the air velocity sensor (4) and correct it.

In VAV systems also, a minimum outside air quantity must normally be supplied for reasons of hygiene. This is why the positioning signal of the flow volume controller has an appropriate low limit.

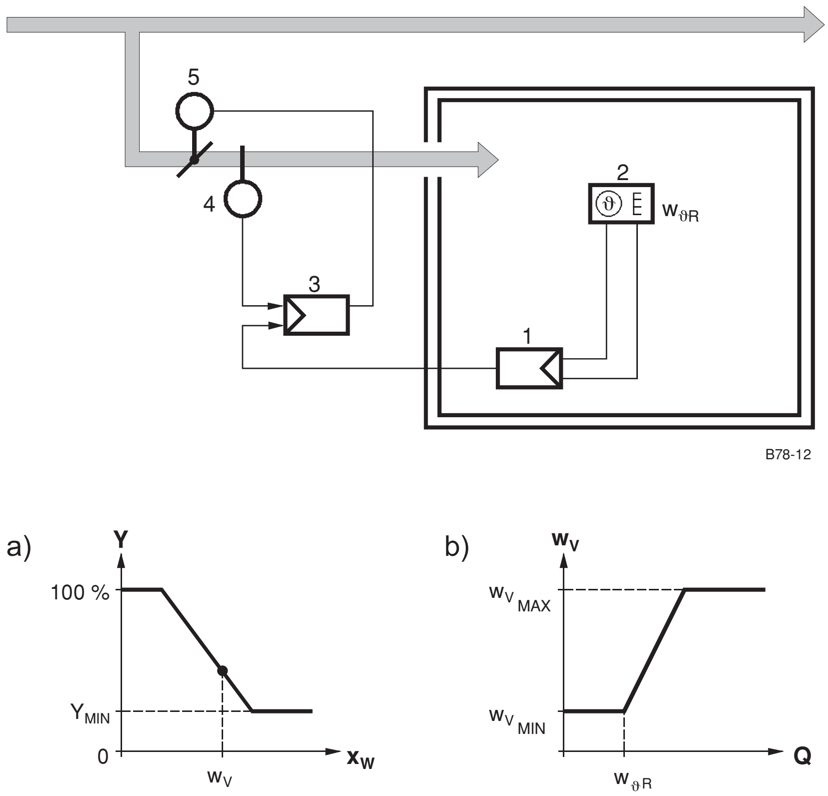


Fig. 8-11 Individual room temperature control with variable air volume (VAV)

1 Room temperature master controller
2 Room unit with temperature sensor and setpoint adjuster
3 Supply air velocity slave controller
4 Air velocity sensor
5 Actuator for air volume controller
a) Function diagram of the air volume controller
b) Function diagram of the room temperature controller
w_V Air volume controller setpoint

VAV controller The supply air velocity controller (3), air velocity sensor (4) and actuator (5) shown in Fig. 8-11 are normally combined in a single piece of equipment, a VAV controller.

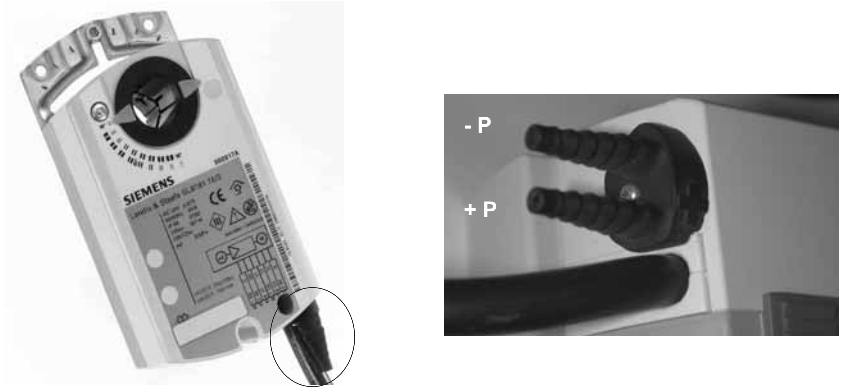


Fig. 8-12 VAV controller with connecting tubes for pressure measurement (detail view on right)
– P Connection for the "lower" pressure of the measuring device
+ P Connection for the "higher" pressure of the measuring device

Frequently, the air velocity sensor is a static pressure sensor (i.e. one without a continuous flow of air) with a silicon diaphragm (4 in Fig. 8-13). This is connected to the VAV unit via measuring lines (3) to the measuring device (2, e.g. a measuring diaphragm or measuring cross). The measured signal is converted in the pressure sensor directly into a measured signal proportional to the volumetric flow rate.

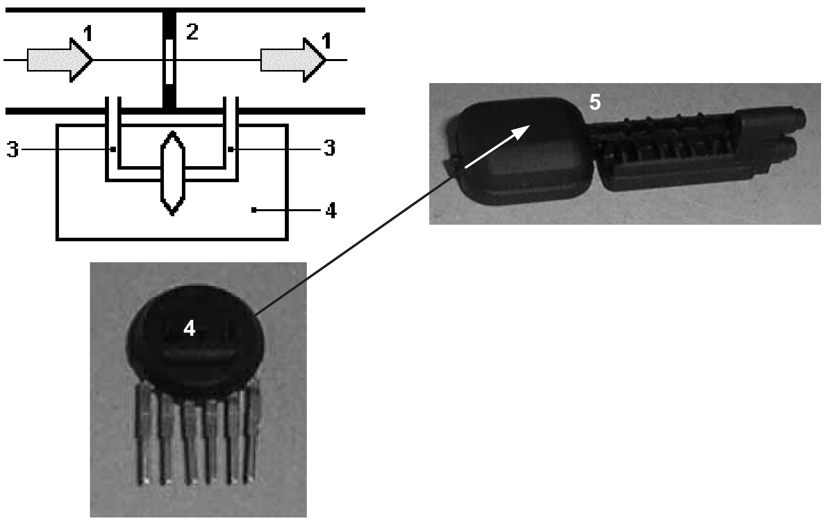


Fig. 8-13 Pressure measurement to determine the flow velocity or air volume velocity
1 Measured air volume
2 Flow resistance in duct (e.g. measuring cross, measuring diaphragm etc,)
3 Measuring lines
4 Static pressure sensor (e.g. silicon diaphragm)
5 Pressure measuring unit with built-in pressure sensor and air valve

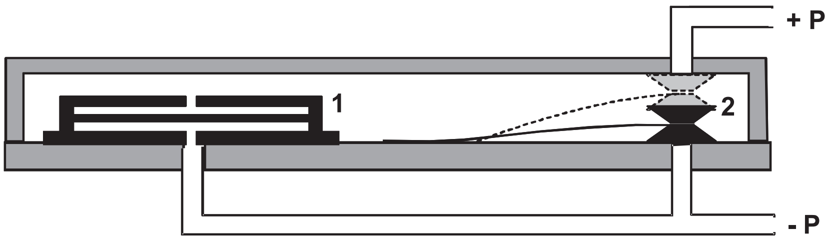


Fig. 8-14 Pressure measuring unit with pressure sensor (1) and air valve (2), arranged for automatic zero-point balancing (the pressure sensor receives the lower pressure "–P" from both sides)

The VAV-unit supplier normally installs the VAV controller on the VAV unit and sets the required air volume (nominal volume flow rate) and the minimum/maximum volume flow rate ($\dot{V}min$, $\dot{V}max$).

Fig. 8-15 VAV controller fitted onto VAV unit

The room temperature controller (Fig. 8-11, 1) transmits the required air volume to the VAV controller in the form of a proportional setpoint signal between the minimum and maximum volume flow rate ($\dot{V}min$... $\dot{V}max$).

If the internal heat in individual rooms is no longer sufficient to maintain the room temperature at the setpoint with this minimum quantity of cooling air, reheaters must be installed locally and controlled in sequence with the flow volume (Fig. 8-16).

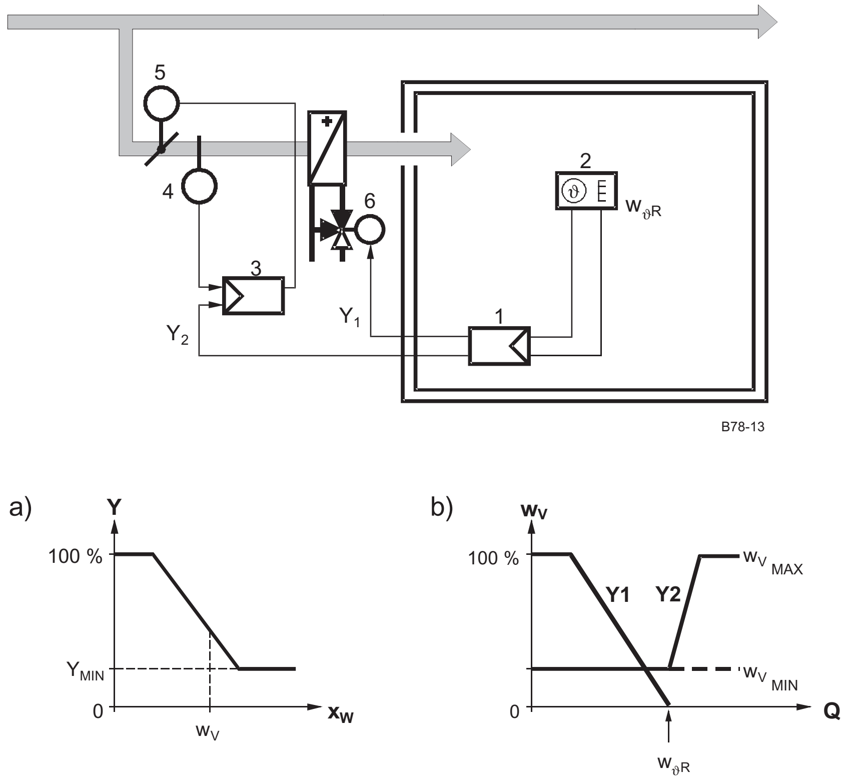


Fig. 8-16 Individual room temperature control with VAV and reheater

1 Room temperature master controller
2 Room unit with temperature sensor and setpoint adjuster
3 Supply air velocity slave controller
4 Air velocity sensor
5 Actuator for air volume controller
6 Actuator for heating valve
a) Function diagram of the air volume controller
b) Function diagram of the room temperature controller
Y_1 Heating valve positioning signal
Y_2 Air volume setpoint compensation
w_V Air volume controller setpoint

8.5.2 Central supply air temperature compensation via the individual room temperature setpoints

In order to prevent too large a difference between the supply air temperature and room temperature in the individual rooms, which would cause drafts, the setpoints of the room temperature controllers can be taken as the reference variable for the supply air temperature. The prerequisite for this control concept (Fig. 8-17) is the communication of the individual room controllers with a building automation and control system via bus.

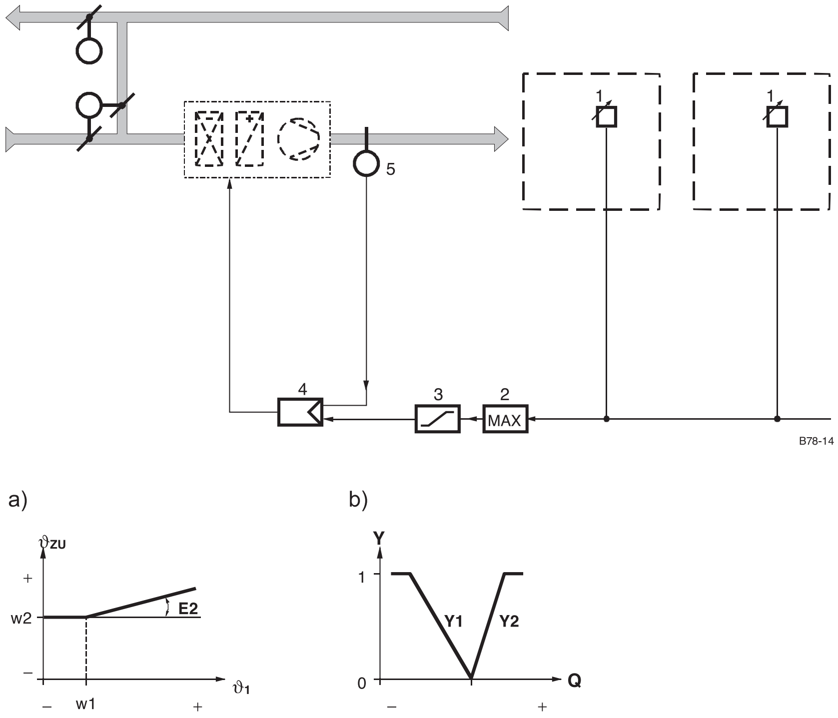


Fig. 8-17 Central supply air temperature compensation in VAV systems
- 1 Room temperature setpoint setting unit
- 2 Determination of maximum or mean value of the room temperature setpoints
- 3 Supply air temperature setpoint shift controller
- 4 Supply air temperature controller
- 5 Supply air temperature sensor
- a) Shift controller function diagram
- b) Function diagram of the supply air temperature controller
- w_1 Basic value of compensation
- w_2 Supply air temperature setpoint

The shift controller (3) generates the setpoint for the supply air temperature controller (4) from the maximum or mean value (2) of the room temperature setpoints (1) according to the set (or programmed) compensation authority (E2). The setpoint is continuously raised if the room temperature exceeds the defined basic value of compensation (w_1).

8.5.3 Primary air volume compensation

The same prerequisites apply to this control concept as listed under 8.5.2. In this case, however, the building automation and control system does not use the setpoints but the current manipulated variables of the room temperature controllers as the reference variable (Fig. 8-18).

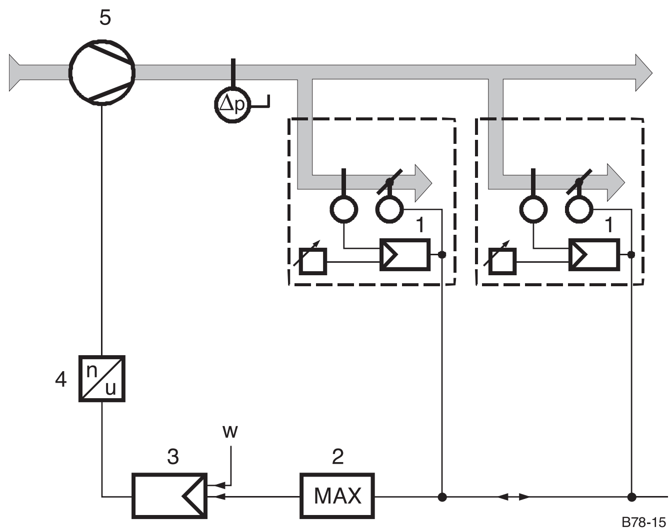


Fig. 8-18 Central supply air volume compensation in VAV systems
1 Room temperature controller positioning signals
2 Maximum selection of the air volume positioning signals
3 Central supply air volume controller
4 Supply air fan speed control unit
5 Supply air fan

The building automation and control system (2) selects the largest air volume positioning signal from all individual room controllers (1) and transfers it as the actual value of the controlled variable to the central supply air volume controller (3). This controller compares the actual value with the programmed setpoint of approximately 90 % of the correcting span of the room temperature controllers. If the current positioning signal falls below the setpoint, the central supply air volume controller reduces the speed of the supply air fan via the control unit (4). This reduces the pressure in the supply air duct, and the room temperature controllers must raise their air volume positioning signals until the largest signal corresponds to the setpoint of the central supply air volume controller (3) again. This prevents the supply air fan from producing too high a pressure, which would have to be throttled back at the room air outlets.

8.5.4 Static pressure control in the air ducts

The throttling of the air outlets via the individual room or zone controllers gives rise to an increase in the static pressure in the distribution duct if the fan speed remains constant. The pressure rise corresponds to the slope of the fan characteristic and can be prevented by controlling the static pressure (Fig. 8-19) in the respective distribution duct. The most effective positioning action of the pressure controller in this case is the speed control of the fans in the central air treatment plant.

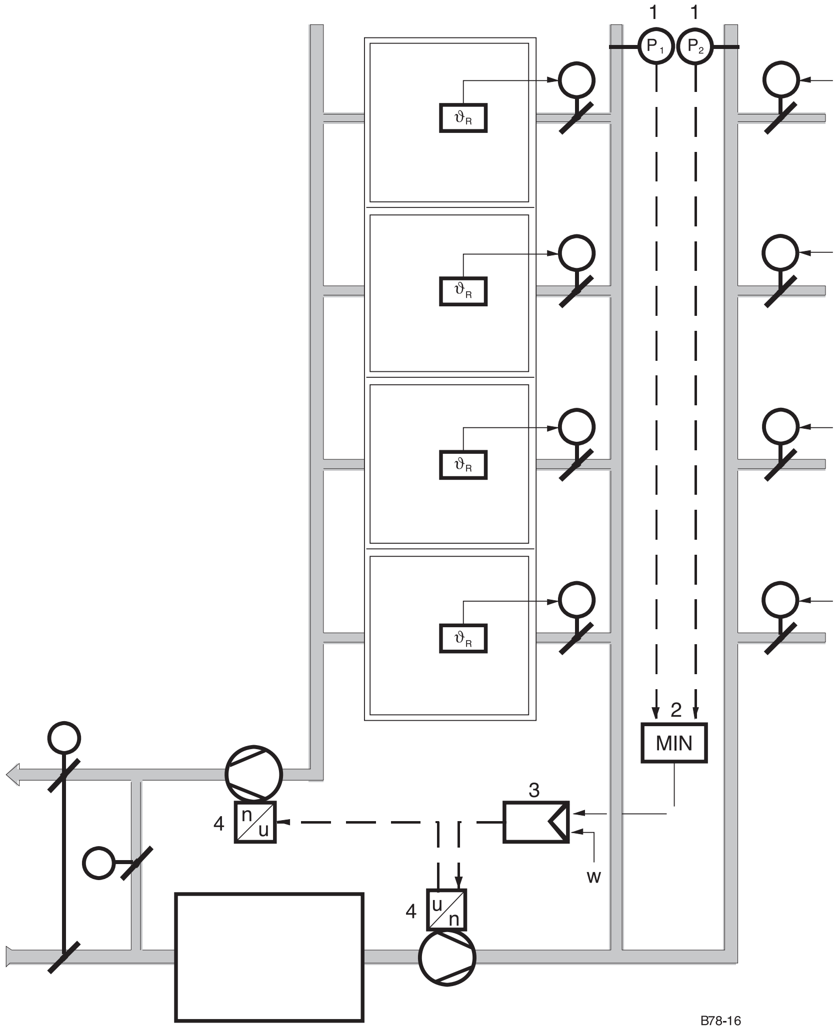


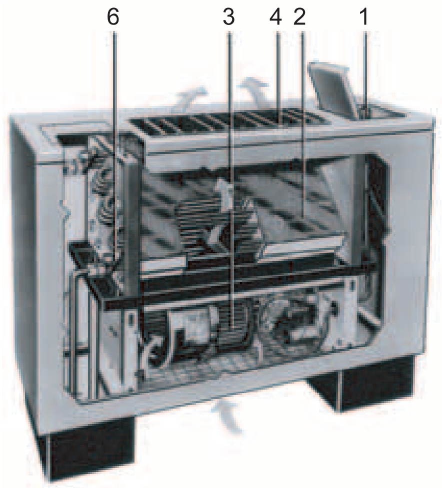
Fig. 8-19 Static pressure control in the air ducts of VAV systems
1 Duct pressure sensor
2 Minimum value selection of the pressure measurement signals
3 Central supply air duct pressure controller
4 Fan speed controllers

The positioning of the duct pressure sensors must be especially well planned. They should be placed at a location where 50 to 70 % of the respective duct resistance can be acquired at the full load air volume. The pressure sensors (1) transmit the measured value to the pressure controller (3) via the minimum value selection (2). The controller compares this value with the defined setpoint and changes the speed of the supply air fan in case of deviation. The speed of the extract air fan is also varied synchronously via a slave control. This solution assumes that the supply air ducting and extract air ducting have approximately the same pressure loss characteristic. If this is not the case, and if the demands regarding stable air conditions in the rooms are higher than average, extract air volume control with compensation for the supply air volume would have to be implemented instead of speed synchronization.

8.6 Fan coil systems

Fan coil systems are ideal air heating and cooling systems for hotel rooms. During heating operation, a central heating system (floor heating) controlled according to the outside temperature provides the base load heating, i.e. the room temperature is maintained at approximately 15 °C in economy mode. On changeover to comfort mode, the fan coil unit achieves the desired comfort temperature within a few minutes. In all other rooms, the fan coil units remain in economy mode or off.

The fan coil unit (Fig. 8-20) is installed on a suitable wall of the room and is connected to the cold and hot water piping and to the electrical supply. If the device is installed on an outside wall, a small quantity of outside air can be drawn in via a manually adjusted damper (5) and mixed with the recirculated air.



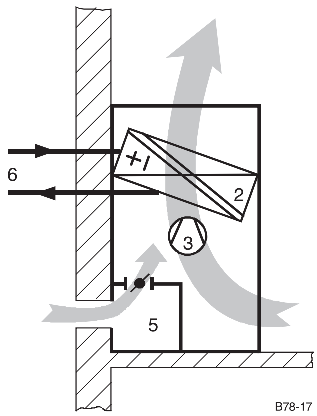


Fig. 8-20 a) Fan coil unit and its components b) Fan coil unit with outside air box

1 Control elements
2 Finned tube heat exchanger
3 Fan
4 Adjustable supply air grille
5 Outside air box with damper
6 Hot or cold water circulation (two-pipe system)

The heating or cooling load in the room or room zone is basically covered by the hot and cold water circulation in a two-pipe or four-pipe system. The two-pipe system can only be centrally switched between heating and cooling operation (changeover system), whereas the four-pipe system provides for individual heating or cooling in each room. Depending on fan coil unit design, air or water side control elements are available for room temperature control. Fig. 8-21 shows the control of a two-pipe fan coil unit with a water-side control element.

The room temperature controller (2) compares the room temperature measured by the sensor (1) against the setpoint. In case of deviation, it adjusts the valve (3). Depending whether hot or cold water is circulating in the supply, the changeover thermostat (4) switches the direction of control action to heating or cooling. If the current direction of control action is heating, the controller opens the valve when the room temperature falls below the setpoint; if the direction of control action is cooling, the controller opens the valve when the room temperature rises above the setpoint (see function diagram, 9). It is possible and useful to set the cooling setpoint 3...4 K higher than the heating setpoint (dead band control, see chapter 1.6).

Room devices also offer the possibility of manual or automatic (presence control) switchover between comfort and economy mode. In economy mode, the heating setpoint is approximately 5 K lower, and the cooling actuation signal is locked out. Additionally, the fan is switched off when the valve is closed.

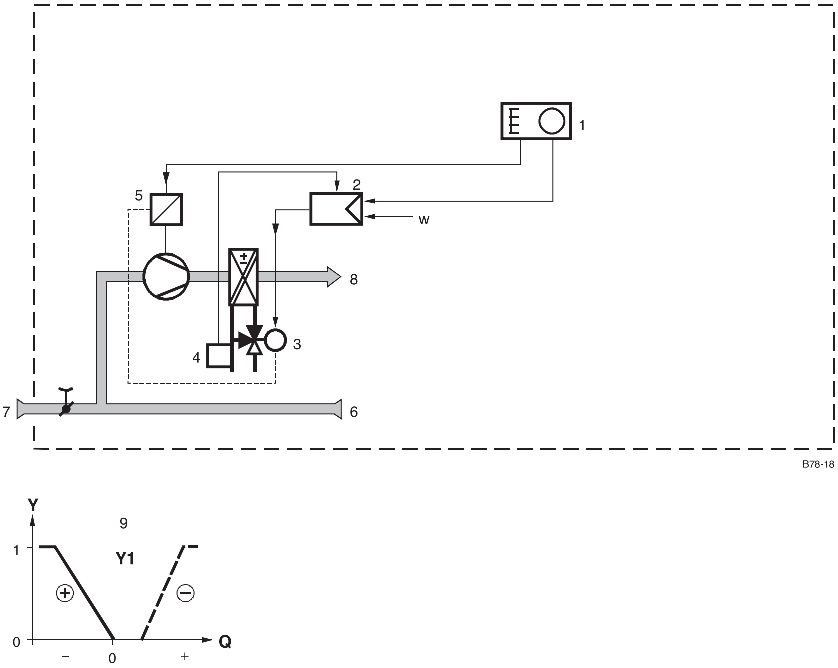


Fig. 8-21 Water-side control of a fan coil unit (two-pipe system)

1 Room unit with temperature sensor and speed selector switch
2 Room temperature controller
3 Actuator for air heating/cooling coil
4 Changeover thermostat
5 Fan speed controller
6 Recirculated air
7 Outside air
8 Supply air
9 Room temperature controller function diagram

In large buildings (hotels), the fan coil units are operated with digital controllers which are managed via bus by a building automation and control system.

This system provides the following functions:
- Definition of normal setpoints for the comfort and economy modes
- Feedforward of setpoint adjustment influences
- Definition of all control parameters
- Comfort mode lockout
- Control function monitoring

8.7 Induction systems

Instead of a fan, induction units contain a noise-absorbing primary air chamber with close-coupled plastic nozzles through which the primary air is blown at high velocity into a mixing chamber, where a negative pressure is created (Fig. 8-22). The negative pressure draws in (induces) room air (so-called secondary air) via a finned tube heat exchanger, where it is heated or cooled as required.

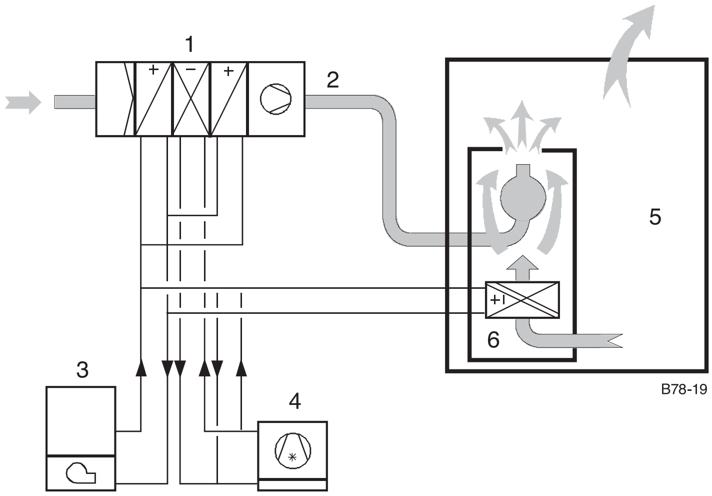


Fig. 8-22 Principle of operation of an induction system
1 Central primary air treatment
2 Primary air duct (high-velocity system)
3 Boiler
4 Water chiller
5 Room
6 Induction unit (two-pipe system)

Induction unit

Fig. 8-23 shows the principle design of an induction unit. The primary air is injected at high velocity into the pressure chamber (4) where the dynamic pressure is converted to static pressure. The static pressure causes the primary air to be distributed evenly to all nozzles and forced into the mixing chamber at high velocity. The injection effect of these air flows gives rise to a negative pressure with respect to the room air, which is drawn in (induced) via the finned tube heat exchanger (6) as secondary air.

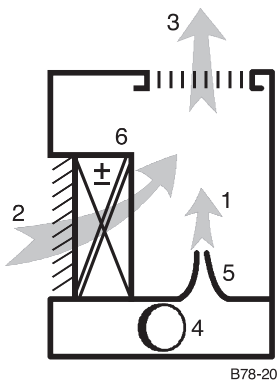


Fig. 8-23 Principle design of an induction unit (two-pipe system)
1 Primary air (treated outside air)
2 Secondary air (room air)
3 Supply air
4 Pressure chamber with Primary air connection
5 Induction nozzles
6 Finned tube heat exchanger

The heat exchanger is supplied with hot or cold water as required. The induced secondary air (2) absorbs the required secondary heating or cooling output and then mixes with the primary air (1). The mixture of secondary and primary air is then blown into the room.

As with fan coil systems, the heating or cooling load arising in the room or room zone is basically covered by the water cycle in a two-pipe or four-pipe system. However, the primary air can provide additional humidification or dehumidification. In order to avoid negative effects on the room temperature control, the primary air is generally supplied at a constant temperature which normally corresponds to the heating setpoint of the room temperature.

The two-pipe system can only be centrally switched between heating and cooling operation (changeover system), whereas the four-pipe system provides for individual heating or cooling in each room. Depending on induction unit design, air or water side control elements are available for room temperature control. Fig. 8-24 shows the control of a two-pipe induction unit with an air side control element.

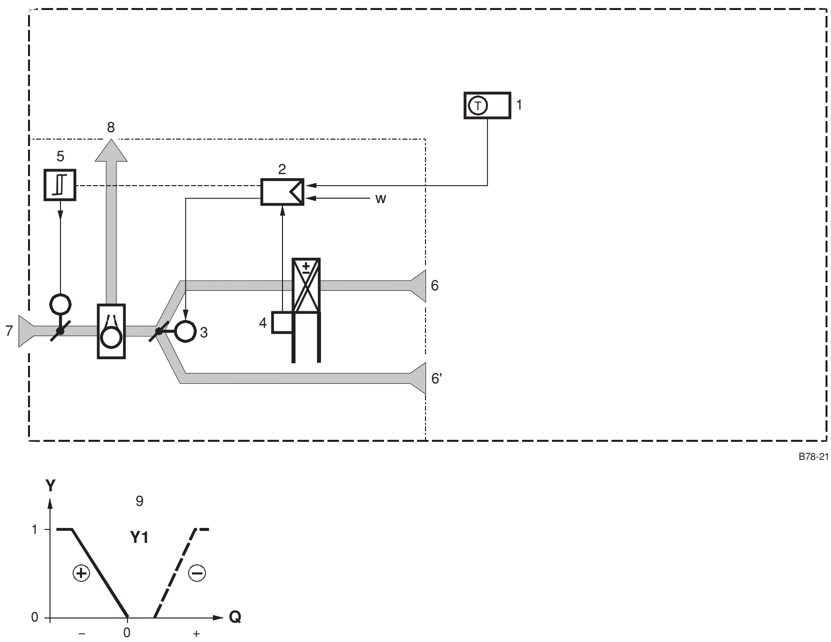
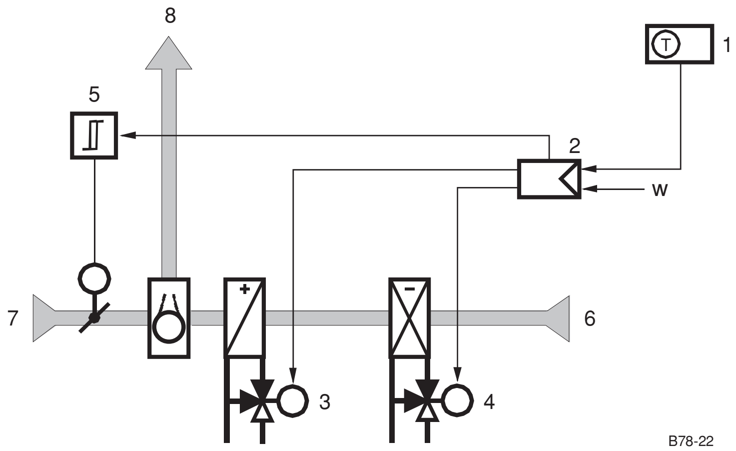


Fig. 8-24 Air side control of an induction unit (two-pipe system)
- 1 Room device with temperature sensor
- 2 Room temperature controller
- 3 Modulating actuator for bypass damper
- 4 Changeover thermostat
- 5 Primary damper ON/OFF controller
- 6 Secondary air flow through heat exchanger
- 6’ Secondary air flow through bypass
- 7 Primary air (treated outside air)
- 8 Supply air
- 9 Room temperature controller function diagram

Two-pipe system control The room temperature controller (2) compares the room temperature acquired by the sensor (1) against the setpoint. In case of deviation, it adjusts the air damper actuator (3). Depending on whether hot or cold water is circulating in the flow, the changeover thermostat (4) switches the direction of control action to heating or cooling. If the current direction of control action is heating, the controller closes the bypass damper when the room temperature falls below the setpoint. If the direction of control action is cooling, the controller closes the bypass damper when the room temperature rises above the setpoint (see function diagram, 9). In economy mode, the primary air damper is closed via the damper controller (5).

Four-pipe system control Four-pipe systems provide for individual heating or cooling in each room. The room temperature controllers have separate positioning signal outputs for heating and cooling, so a changeover thermostat is not needed. Depending on induction unit design, air-or water-side control elements are available for room temperature control. Fig. 8-25 shows the control of a four-pipe induction unit with water-side actuating devices.



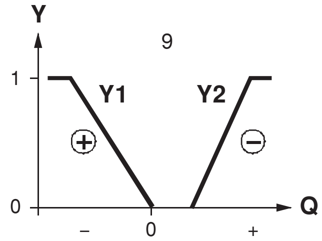


Fig. 8-25 Water-side control of an induction unit (four-pipe system)

1 Room unit with temperature sensor
2 Room temperature controller
3 Modulating actuator for heating valve
4 Modulating actuator for cooling valve
5 Primary damper on/off controller
6 Secondary air flow through heat exchanger
7 Primary air (treated outside air)
8 Supply air
9 Room temperature controller function diagram
Y_1 Positioning signal heating valve
Y_2 Positioning signal cooling valve

The room temperature controller (2) compares the room temperature acquired by the sensor (1) against the setpoint. In case of deviation below the heating setpoint, the controller opens the heating valve (3). In case of a deviation above the cooling setpoint (= heating setpoint + dead zone), it opens the cooling valve (see function diagram, 9). In economy mode, the primary air damper is closed via the damper controller (5).

Otherwise, the control and monitoring functions, such as dead zone control, comfort and economy mode, are identical to those of fan coil units (see 8.6), with or without a building automation system.

8.8 Flow temperature setpoint compensation

In fan coil or induction systems in which room air conditioning devices are supplied with centrally treated hot and cold water, it can be useful to adjust the flow temperatures to the load state of these air heating and cooling devices. The current positioning signals of the room temperature controllers can be taken as the reference variable. The prerequisite for this control concept (Fig. 8-26) is the communication of the individual room controllers with a building automation and control system via bus.

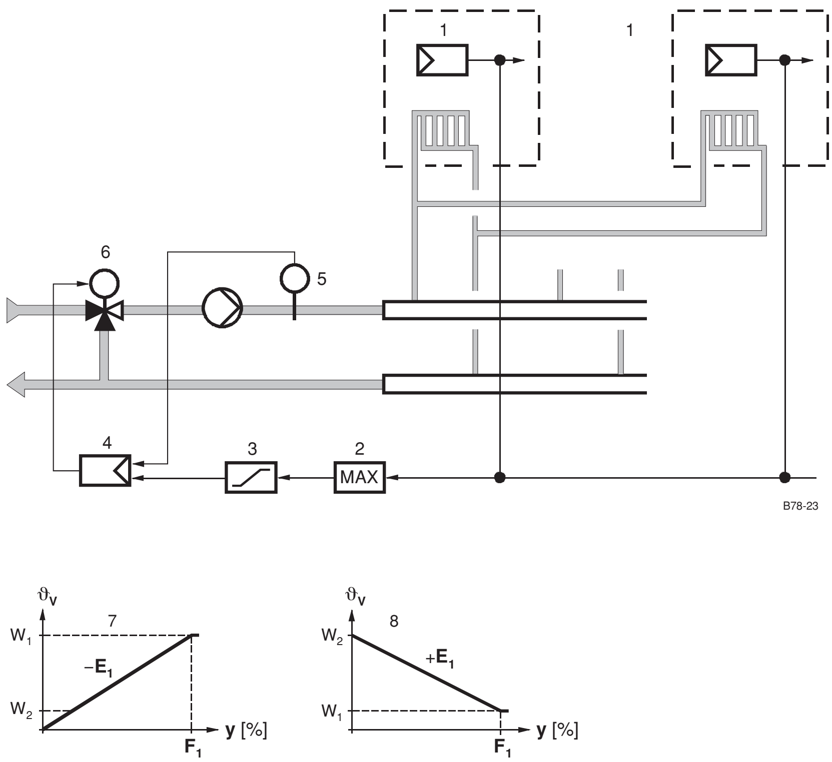


Fig. 8-26 Flow temperature compensation for fan coil and induction systems
1 Room temperature controller
2 Determination of maximum or mean value of the room temperature positioning signals
3 Flow temperature setpoint shift controller
4 Flow temperature controller
5 Flow temperature sensor
6 Flow temperature control element
7 Shift controller heating function diagram
8 Shift controller cooling function diagram
F_1 Basic value of compensation (e.g. y = 90 %)
w_1 Setpoint for full load operation

The building automation system selects the currently largest positioning signal ((6), heating or cooling) from the connected individual room controllers and transmits it to the series-connected setpoint shift controller. If the largest positioning signal with the heating direction of action is smaller than the adjusted basic value of compensation F1 (e.g. y = 90 %), the continuous lowering (function diagram (7)) of the hot water flow temperature setpoint begins according to the programmed compensation authority –E1. If the largest positioning signal with the cooling direction of action is smaller than the adjusted basic value of compensation F1 (e.g. y = 90 %), the continuous raising (function diagram (8)) of the cold water flow temperature setpoint begins according to the programmed compensation authority +E1.

Depending on the different load behaviors of the individual rooms, the mean value of all positioning signals could be taken as the reference variable instead of the maximum positioning signal.

8.9 Compact room air-conditioners

Compact room air-conditioners are used for cooling (and partial dehumidification) or heating of the room air. They are provided by the manufacturer with the necessary control and safety equipment and supplied as "ready-to-connect" devices. This group of devices includes:

- window units
- floor-standing units
- cabinet units
- split air-conditioners

8.9.1 Window air-conditioning units

In these devices, the fan speed as well as the heating and cooling output can be manually adjusted between maximum and minimum levels. In the case of window units that provide heating and cooling (heat pump devices), the switchover from heating to cooling operation is accomplished by reversing the flow of the refrigerant via a four-way valve. This exchanges the evaporator and condenser functions. The thermodynamics of the refrigeration cycle are explained in the training module B01 "Introduction to HVAC services," chapter 4 "Basic physical principles."

8.9.2 Floor-standing air-conditioning units

Floor-standing air-conditioning units fulfill the same functions as window units and are equipped with the same control devices.

8.9.3 Split air-conditioners

The functional design of a split air-conditioner (Fig. 8-27) basically consists of a refrigeration cycle with a compressor (3), an air-cooled condenser (4), an expansion valve (5) and a direct expansion air cooler (2). An air heater could also be installed as an additional component in the air circulation unit if the air cooler is also to be used for dehumidification.

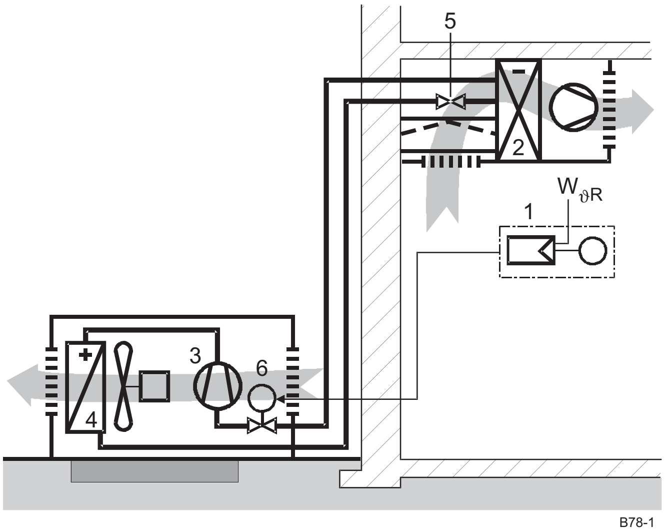


Fig. 8-27 Design of a split air-conditioner
1 Room temperature controller
2 Air cooler (evaporator)
3 Compressor
4 Condenser (air-cooled)
5 Expansion valve
6 Suction valve

Room temperature (1) control is achieved by acting on the refrigerant cycle via a suction valve. This allows the cooling output to be adjusted to the demand by modulation between approximately 40 % and 100 % and by ON/OFF control below 40 %. If an electric air heater is also installed, its output must be controlled by an additional heating sequence of the controller via a step switch or current valve.

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Siemens Switzerland Ltd
Building Technologies Group
International Headquarters
Gubelstrasse 22
CH-6301 Zug
Tel +41 41 724 24 24
Fax +41 41 724 35 22

Siemens Building Technologies Ltd
Hawthorne Road
Staines
Middlesex TW18 3AY
United Kingdom
Tel +44 1784 46 16 16
Fax +44 1784 46 46 46

Siemens Ltd
Building Technologies
Units 1006-10
10/F, China Resources Building
26 Harbour Road
HK-Wanchai
Tel +852 2870 7888
Fax +852 2870 7880

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